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SUNFLOWER TURBOALTERNATOR CSU I-3A

4329-HOUR-TEST SUMMARY REPORT

Prepared under Contract No. NAS 5-462 *by*
THOMPSON RAMO WOOLDRIDGE, INC.
Cleveland, Ohio

for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • APRIL 1965

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TABLE OF CONTENTS

1.0 INTRODUCTION

1.1 Scope

Testing of Sunflower CSU I-3A Rankine Cycle turboalternator consisted of three separate runs of 769, 3556 and 4 hours. The first run began on February 5, 1963 and continued until March 9, 1963 when it was voluntarily terminated because of blockage of the first stage turbine nozzle. The unit was partially disassembled and discrete particles which blocked the first stage nozzle were removed. No other changes were made to the unit and after its reassembly a second test run was begun on April 16, 1963. This test continued until September 11, 1963 when a municipal power failure forced a shut down of the unit. It sustained no apparent damage due to the shut down and, after minor rig repairs, was restarted for the third test run which was terminated when the rig side alternator connector shorted. The connector short resulted in demagnetization of the alternator stator bore seal, overspeed, and ultimate shaft seizure.

This report covers the significant factors related to performance and endurance capabilities of the CSU I-3A, including design factors, results of testing, performance analysis, and post-test hardware inspection. The report also includes CSU disassembly notes, a discussion on the cause of failure, and recommendations for developmental improvements.

1.2 Test Objectives

Objectives for CSU I-3A testing were outlined in the test plan as follows:

A. Obtain design point operation.

B. Determine pump performance characteristics.

C. Determine various drain flow rates at the design point.

   1. Turbine bearing drain
   2. Alternator-thrust bearing drain
   3. Alternator cavity drain
   4. Turbine vapor by-pass line

D. Determine effect of varying bearing lube flow rates on drain flow rates, bearing clearances, bearing whirl, and other CSU parameters at design lube temperature.

E. Impose a 12" Hg head (above CSU ℄) on the bearing drains to determine the effect of separately and simultaneously "flooded" bearings on:

   1. Alternator cavity drain flow
   2. Turbine vapor by-pass line flow
   3. Bearing clearances
   4. CSU output power (shaft drag)
   5. Local bearing support, housing, and drain temperatures

F. At nominal bearing lube flow rates vary all lube temperatures together to 350, 375, and 450°F to evaluate effect on bearing clearance and other CSU parameters.

G. At design point operation, vary alternator coolant temperature to 525, 500, and 475°F to determine effect on alternator cavity drain flow rate (condensation), temperature, and on electrical output performance.

H. Shut down the CSU and conduct a simulated squirt start (or starts) with the objective of accelerating to 40,000 rpm at design cycle flow under simulated PCS boiler output conditions.

I. Gain preliminary evaluation of corrosion product trapping devices and rig cleaning in terms of endurance testing consistent with other test schedules and hardware availability.

1.3 CSU Description

The CSU (combined shaft unit) is a mercury Rankine cycle turboalternator with turbine, condensate pump, and alternator mounted on the same shaft. Figure 1 is a cross sectional view of the CSU I-3A model which was tested for 4329 hours. Figure 1A illustrates a more advanced model, CSU I-2A, which incorporates features required for orbital flight operation. The turbine is a three-stage axial flow impulse type, with partial admission first and second stages and full admission third stage. The alternator is a wound stator, permanent magnet rotor type, six pole, two phase 2000 cps machine. The alternator is cooled by the boiler inlet flow and therefore acts as a boiler preheat source. The condensate pump supplies bearing flow as well as mercury cycle flow. This pump is a jet boost centrifugal impeller type, with the impeller

SUNFLOWER TURBO-ALTERNATOR

(CSU-I-3A)

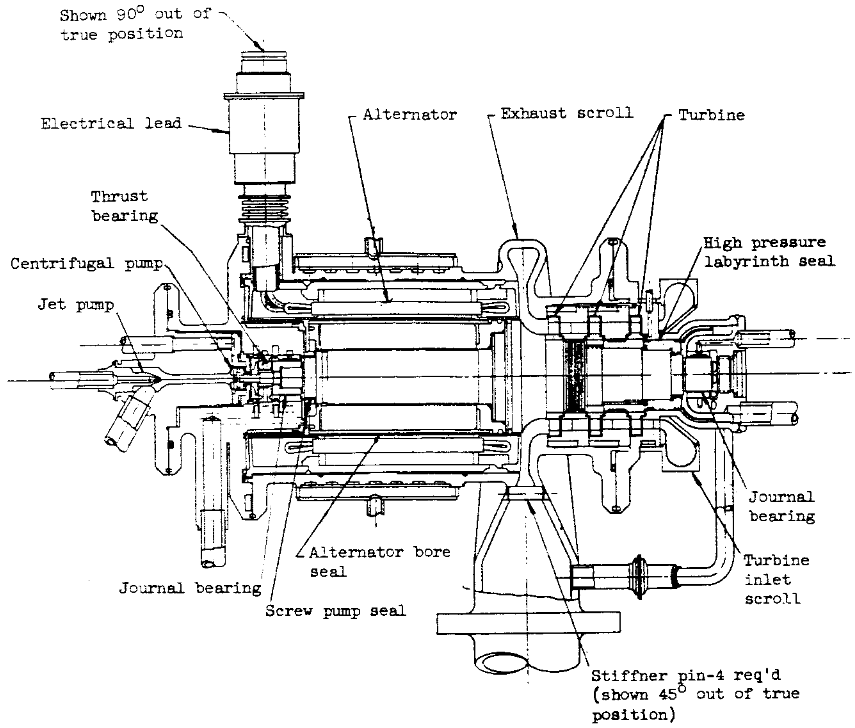


FIGURE 1

SUNFLOWER TURBO-ALTERNATOR

(CSU-I-2A)

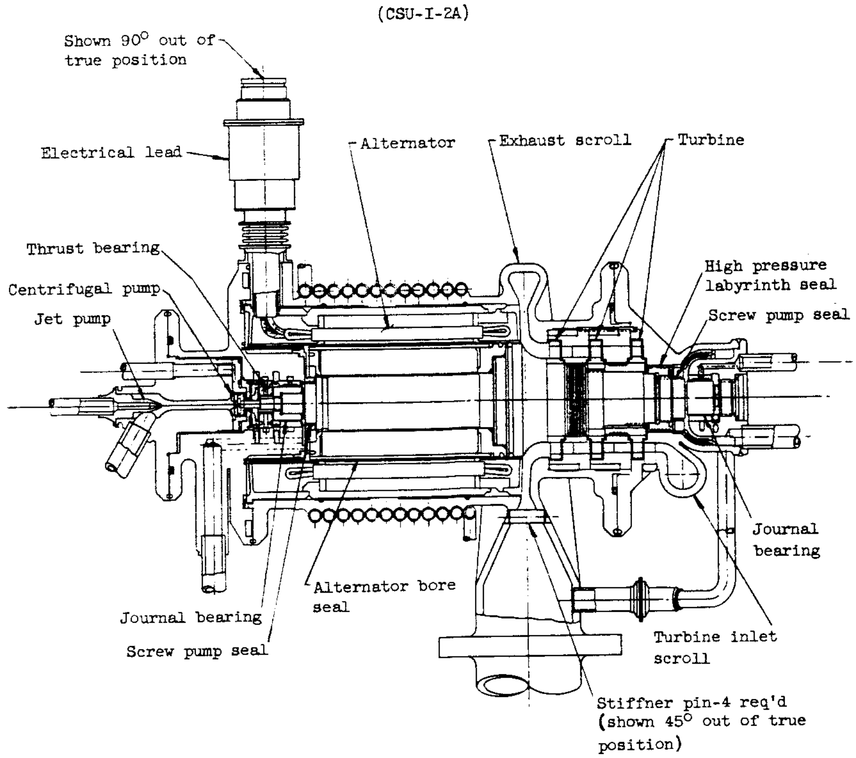


FIGURE 1A

mounted on the end of the turbine alternator shaft, and therefore operates at the shaft speed of 40,000 rpm.

The complete shaft assembly is supported by two journal bearings and a double acting thrust bearing. The two journals are sleeve type rotating members supported by three sector type bearings. Each sector comprises a lube feed cavity and an axial lube drain slot. The thrust bearing contains spiral grooves in the thrust faces and an annular orifice which combine to produce a self-centering feature. The mating faces to the bearing are plane faces discs. All three bearings are lubricated with liquid mercury from the condensate pump.

The bearings, alternator stator, turbine nozzles, pump casing, and various connecting lines are all contained and supported by a housing assembly which also comprises the turbine exhaust passage and mounting support for the CSU. Housing assembly and line connections are all provided with weldable flanges to allow hermetic sealing of the complete assembly itself or as a part of a system.

CSU I-3A was a rebuild unit nearly identical to CSU I-3 (and CSU I-1A) and contained the following parts:

a. Shaft - the shaft was originally built for CSU I-2, but was reworked to the CSU I-3 design specifications. This shaft had a welded pump shaft extension which resulted from a previous fabrication deviation.
b. Turbine Inlet Housing - this housing was used on the CSU I-1 test but was reworked to the CSU I-3 design specifications (wrap-around scroll, etc.) and contained a new turbine bearing bushing. Due to necessary refacing, this housing required a shim between it and the alternator housing to retain proper axial locations.
c. Alternator Housing, Alternator Stator, and Alternator Bearing Support Housing - these elements comprise a welded assembly which was used in the CSU I-3 2350 hour test. This assembly was used in CSU I-3A with no rework or repairs other than cleaning. This means that the alternator bearing, thrust bearing, filters, stator and housing pilot surfaces were used in the same condition left from the endurance test. The thrust

bearing was in excellent condition as was the journal bearing, except for one or two very tiny surface scratches. The alternator stator bore seal exhibited a tiny surface crack which exceeded the allowable helium leak rate. As a precautionary measure nitrogen was used as a cover gas in the stator to prevent the entrance of mercury.

d. Jet Pump Housing - this housing was the original CSU I-3 housing, the only revisions being that a filter was added in the jet supply line and the jet nozzle replaced. The pump volute and venturi have been retained as left from the endurance test.

e. Second and Third Stage Turbine Nozzles - the second stage nozzle was built for CSU I-2 and was used as is. The third stage nozzle was built for and used in CSU I-1 test. However, it had an excessive throat area and therefore, six nozzle passages were blocked to obtain the design throat area.

The CSU was installed and tested in a performance test rig. This test rig was capable of supplying and controlling all of the CSU input and output mercury parameters to simulate design and off-design system impositions on the CSU. In addition, the test rig contained a manually adjustable simulated electrical load and a parasitic load speed control similar to the system prototype controller. The test rig contained instrumentation to measure and/or record all parameters of temperature, pressure, and flow rate as well as vibration and alternator electrical output (frequency, power, current, and voltage). It also had provisions for measuring bearing drain flows, alternator cavity drain flow, and turbine vapor by-pass line flow.

1.4 CSU Specifications

Nominal operating specifications applicable to current CSU's are as follows:

1.4.1 Power Output

a. Useful	3.0 Kw
b. Speed control	.150
c. Allowance for performance degradation	.350
Total	3.5 Kw

1.4.2	Voltage and Frequency	
	a. Voltage	110 AC Volts
	b. Frequency	2000 cps
	c. Phases	2
1.4.3	Regulation and Distortion	
	a. Voltage	110±5 Volts
	b. Frequency	2000±20 cps
	c. Distortion (harmonic)	Total content - less than 7% of fundamental
1.4.4	Power Factor	0.8 lagging to unity
1.4.5	Operating Life	1 year unattended
1.4.6	Turbine Parameters	
	a. Inlet pressure	240±10 psia
	b. Inlet temperature	1250 + 50 -175°F
	c. Inlet flow	13.7±1 lb/min
	d. Exhaust pressure	7.0±.5 psia
1.4.7	Journal Bearings	
	a. Supply flow (each)	6.5±1.5 lb/min
	b. Supply temperature	399±6 F
1.4.8	Thrust Bearing	
	a. Supply flow	8.0±2 lb/min
	b. Supply temperature	399±6 F
1.4.9	Mercury Pump	
	a. Output flow	34.7±6 lb/min
	b. Output pressure	515 psia (min)
	c. NPSH	not less than 3.4 psia
	d. Inlet temperature	393±6 F

2.0 SUMMARY OF RESULTS

In general all of the stated objectives established at the outset of testing were fulfilled with only one notable exception. Simulated squirt starts, scheduled to be made at the completion of the endurance run, were precluded by the unexpected failure. The total accumulated run time of 4329 hours demonstrated an endurance capability of half its design life. Also the following factors indicate that the unit could reasonably have been expected to complete the endurance run:

a. The primary cause of failure was independent of the CSU.

b. There was no measurable deterioration in performance up to the time of failure.

c. Post-test dimensional and metallurgical examination of the hardware did not disclose any areas of imminent failure.

There are, of course, some questionable areas such as erosion of the pump impeller and the turbine and creep of the rotor shrink sleeve but considering the severity of the failure mode, the hardware remained in good condition.

Bearing performance was an important aspect of the CSU I-3A testing in view of the fact that the thrust and alternator bearing stationary members began the test runs with 2348 hours running time already accumulated. The forces of shut down caused some damage to the journal bearings, especially in the case of the alternator bearing. However, the only change attributable to normal operation was an occasional .85 mils of corrosion. In the case of the thrust bearing there was no damage whatsoever; it was literally "good as new". No signs of wear, erosion, or dimensional changes could be found as a result of the operational test exposure.

Although CSU I-3A is very nearly identical to CSU I-3, which ran for 2348 hours in 1962, the CSU I-3A test results are more enlightening in some respects because of two significant test rig modifications. One was the incorporation of corrosion product trapping devices which virtually eliminated the corrosion product accumulation which caused the CSU I-3 failure. The other was the addition of provisions for measuring bearing drain flows and alternator and turbine cavity drain flows. This made possible a more realistic appraisal of CSU performance as the test progressed.

The results of the CSU I-3A testing verified the findings of the CSU I-3 test. Both test series indicated that the basic CSU design is capable of meeting its design performance requirements but that further design refinements are required in certain areas. These refinements would not involve any changes in CSU operational parameters, nor would they require extensive changes in the basic layout. In fact, most of the desired changes have been integrated into later units using existing semi-finished hardware, with little or no modifications to the original major subcomponent design. Further discussion of these developmental modifications is presented in a later section of this report.

3.0 DEVELOPMENTAL HISTORY

This section presents a very brief history of developmental test and re-design work which preceeded the CSU I-3A test runs.

3.1 CSU I-1 Test Results and Conclusions

The first Sunflower turboalternator design to be built and tested was CSU I-1. This unit was placed on test November 22, 1961. The test lasted only about 1 1/2 hours, but was very illuminating relative to the first design. On first start, the unit accelerated to 25,000 rpm quite freely. It subsequently appeared to be erratic in both speed and power, suggesting liquid drag on the shaft. This condition prevailed throughout the test. The speed was ultimately increased to 40,000 rpm, but alarming variations in power combined with apparent increases in bearing clearances (from flow-pressure data) suggested an impending failure. Therefore, the CSU was shut down to allow examination of data and hardware. Post-test bearing calibrations verified that bearing wear had taken place and the CSU was therefore removed from the test rig for disassembly and inspection.

Inspection revealed extensive wear on the alternator journal and thrust bearings and light wear on the turbine journal. Examination of the bearing drain lines showed severe flow restrictions at several of the butt welds.

Data and performance analysis conclusions were as follows:
Bearing wear resulted from (1) excessive loads and (2) reduced load capacity of the bearings.

1. Excessive shaft loads were caused by:
   a. The presence of liquid Hg around the shaft due to restricted drain lines plus possibility of liquid Hg in the alternator rotor gap.
   b. A large magnetic rotating load due to rotor magnetic imbalance.
   c. A pressurized turbine bearing drain collector (causing excessive thrust) due to vaporization of lube drain fluid which was restricted from draining.

2. Reduced load capacity of the bearings was caused by:
   a. The alternator journal bearing showed evidence during test of thermal increases in clearance between the housing and bushing prior to the wear indications, which would have weakened the bearing capacity. However, the wear patterns on both journals indicated excellent operating alignment so that misalignment was not a factor in reduced load capacity for the journals.
   b. The thrust bearing wear pattern indicated misalignment which resulted from eccentric static loading by its retaining device, thereby reducing its capacity.

3.2 Developmental Redesign for CSU I-3

The CSU redesign was aimed at control of package thermal and stress distributions to maintain proper bearing clearances and alignments and to prevent condensation in vapor spaces around the rotor. Redesign was also directed to reduce the magnetic shaft imbalance by improved quality control, elimination of drain flow passage restrictions, and improved mechanical support of the bearings.

A thermal stress computer analysis was conducted on the bearings and supports. The results were used to specify the assembly fits and clearances required to insure proper and safe operating fits and clearances. Experimental bench tests and thermal analysis of the alternator were conducted to predict coolant temperatures required for proper rotor gap conditions. These analyses and the results of the CSU I-1 test were used to redesign the next available unit (CSU I-3). Each step of the redesign was a logical attack on specific problems evidenced from the first test, but within the framework of the original concept and hardware. The changes involved are listed below. Most of these changes are illustrated in Figure 1. Figure 1 is a slightly more advanced design than CSU I-3 and therefore the alternator cooler, turbine bearing seal configuration, and exhaust scroll struts are additions. CSU I-3 details in these specific areas were left identical to CSU I-1.

1. The turbine inlet was changed from a ducted direct entry design to a 360° collector scroll to improve thermal and mechanical symmetry relative

to the turbine bearing alignment. Misalignment was not a demonstrated problem in CSU I-1 test, but test conditions were not completely representative in this regard.

2. Revisions were made in the alternator bearing housing to provide better thermal separation from the alternator housing and improve thermal separation from the alternator housing and improve thermal and mechanical symmetry relative to journal and thrust bearing alignment. Bearing feed and drain bosses were slit away from the flange and a Supramica washer inserted between flanges for thermal separation.

3. The thrust bearing retainer axial loading was reduced and made symmetrical by modification of associated static seals.

4. The alternator journal bearing bushing fit was increased slightly to provide a better lube seal, and cold bearing diametral clearance was reduced from .0018 to .00105 in. in accordance with computer thermal analysis. The target for operating clearance was .0014 - .0015 inches.

5. An evacuated heat shield was placed around the alternator bearing support barrel to help control bearing clearance, to keep the support cool, and to reduce condensation in the alternator rotor cavity.

6. The thrust bearing lube supply annulus was placed in the housing rather than the bearing O.D. to provide a better lube seal, improved mechanical support, and bearing strength.

7. Axial thermal take-up provisions on the shaft were provided by incorporating belleville springs under the turbine locknut. The thrust spacer material was also changed. These modifications were made to improve shaft integrity and balance.

8. Quality control on magnetic rotor imbalance resulted in a shaft imbalance

of 2-4 lb for CSU I-3 compared 18.5 lb estimated for CSU I-1.

9. The turbine journal sleeve support lands were eliminated and the sleeve was placed in full contact with the shaft to improve mechanical support.

10. The turbine bearing clearance was reduced from .0018 to .00135 in. in accordance with computer thermal analysis. The target clearance was .0011 in. to .0016 in., so that the assembly clearance was a compromise depending on whether or not seal vapor entered the drain cavity.

11. High pressure labyrinth seal and heat shield diametral clearances were reduced along with bearing clearances to improve turbine performance and reduce temperature effects on the turbine bearing.

12. X-ray quality control was applied to all tubing welds to avoid butt weld penetrations into the flow passages.

13. The cavity between alternator stator and housing was sealed at one end with a belleville washer and at the other with a close clearance fit. Sealing this gap provides three functions:

    a. It keeps liquid Hg out of the rotor cavity to avoid liquid binding.
    b. It affords symmetrical heat transfer of alternator loss heat to the coolant by keeping liquid Hg in the stator O.D. gap.
    c. It affords better circumferential temperature distribution and consequently better bearing alignments.

14. A second alternator garter spring support was added to eliminate resonant vibrations which could affect the alternator bearing loads or possibly crack the glass bore seal.

15. The alternator housing and bearing support flanges were Electron Beam welded rather than Heliarc to preclude weld distortions and stresses and afford better bearing alignment.

The list of changes was extensive, but was largely associated with details based on a better knowledge of the unit from results of the first test. The basic concept was unchanged, no new castings were involved, and nearly every change was applied to semi-finished hardware existing from the previous design.

3.3 CSU I-3 Test Results and Conclusions

The CSU I-3-1 test was largely endurance running at various levels of speed and power with most other parameters constant. The discussion of this test is divided into five categories; prestart, start-up and speed climb, 35,000 rpm endurance run, 40,000 rpm endurance run, shut down, and conclusions. More detailed information on this test can be found in TRW ER-5432, Revision A, "Sunflower Turboalternator CSU I-3 2348 Hour Test Summary Report".

3.3.1 Pre-Start

During pre-start calibrations an undersized valve in the drain line caused liquid mercury to back-up into the bearing drain areas within the CSU. In the CSU I-1 test this condition quite likey flooded other areas such as the alternator rotor gap and turbine wheel wells during start-up. Combined with the CSU I-1 drain restrictions and shaft imbalance noted earlier, the rig drain problem without doubt augmented the circumstances of its failure. The drain system was revised and the various static calibrations were completed quite satisfactorily without further incident.

3.3.2 Start-Up and Speed Climb

At 50 psia turbine inlet pressure and full manual load setting the unit speed quickly climbed to 15,000 rpm and after a few minutes stabilized at 20,000 rpm. After 2 hours the speed was raised to 25,000 rpm, held there for 3 hours, and then was raised to 35,000 rpm. The speed was held at 35,000 rpm for half an hour, increased to 40,000 rpm for about one hour, and then returned to 35,000 rpm. During this speed climb the unit performed as expected with two factors of particular interest. First mode critical shaft speed was predicted at 27,000 to 33,000 rpm. It occurs when the shaft mass and bearing lube film "spring rate" act as a resonant system with neligible deflections in the shaft itself. Vibration accelerometers confirmed this condition at the 30,000 rpm plateau and therefor dwell time at 30,000 rpm was kept very brief. This particular critical speed was not considered detrimental but simply undesirable in a new unit with no test history at such a condition.

The second factor of interest is the lower power output at 40,000 rpm compared to power at 35,000 rpm with the same turbine inlet pressure. This condition was expected from predicted performance data. It happens because in this speed and power range, an increase in speed will decrease alternator efficiency and increase shaft parasitic loads (bearings, pump seals, etc.) However, turbine efficiency is this speed range does not increase with speed nearly as rapidly as the parasitic demands; therfore, power is diverted to losses and net electrical power decreases. In general, the output power was less than the predicted performance at these conditions by 100 to 150 watts. Turbine inlet pressure was not brought up to design level during this speed climb time period.

The CSU package pump performed in excess of its design requirements during this period of testing. While operating at 40,000 rpm and supplying its own jet boost flow it was pumping 42 ppm Hg at 500 psig which is 20% higher than nominal design flow and pressure. Bearing operation was quite satisfactory over the full range of speeds. However, at 30,000 rpm a trace of half speed whirl was noted along with the first mode critical speed noted earlier. As speed and power were increased the alternator bearing diametral clearance increased with speed up to 20,000 rpm and did not change with further speed increases. The double acting thrust bearing supply pressure increased between 20,000 rpm but returned to its original value at 40,000 rpm. This was attributed mainly to the pumping action of the spiral grooves being less effective at speeds above 25,000 rpm because counteracting centrifugal forces.

3.3.3 35,000 RPM Endurance Test

An objective of the CSU I-3-1 test was to run for a minimum 500 hours continuous at a confident level of operation as near to design as possible. Three areas of concern led to the decision to make this run at 35,000 instead of 40,000 rpm. These factors were:

a. Half speed bearing shirl.
b. Half speed and shaft speed vibration levels in excess of 1 "g".
c. Possible overstressing of the alternator rotor sleeve due to rotational stress.

Although it had not been established that any of these three items would definitely lead to CSU failure all three could be eased by a reduction in speed and thus the run was made at 35,000 rpm, the next lower speed control frequency.

Alternator power and vibration levels at constant turbine inlet pressure were very stable through the first half of the run, so that power increases were made as planned up to 2,500 watts total near 500 hours. At this power the turbine inlet pressure was 200 psia. In general, operation was quite steady. The journal bearings estimated diametral clearances remained essentially constant as did the vibration levels at full and half shaft speeds. The thrust bearing supply pressure increased with increasing turbine inlet pressure, however, indicating a change in the axial position of the shaft which would affect the effective flow area. Pump operation was very steady for the first 325 hours when the jet nozzle apparently fouled which reduced pump performance.

3.3.4 <u>40,000 RPM Endurance Test (500 to 2348 Hours)</u>

As planned at the outset of the 35,000 rpm, 500 hour test run, the bearing whirl, vibration amplitude, and alternator rotor sleeve stress concerns were all reviewed prior to initiating the 40,000 rpm test run. Half speed bearing whirl showed no signs of deteriorating bearing performance based on accelerometer data and assessments of running bearing clearances. It was decided that the measured vibration amplitudes were not excessive; the shaft speed amplitude in the horizontal direction was, in fact, zero. A re-evaluation of the factors affecting alternator rotor sleeve temperature disclosed a safer operating condition than originally determined.

When the speed was raised to 40,000 rpm, operating conditions changed generally in the manner that was expected. There was a slight reduction in alternator power due to changes in shaft parasitic loads and alternator efficiency as previously explained. The thrust bearing supply pressure changed because of the greater centrifugal forces acting on the fluid in opposition to the pumping characteristics of the bearing. There were increases in vibration levels, journal bearing supply pressures, and other parameters such as voltage and pump discharge pressure.

Shortly after achieving 40,000 rpm, the turbine inlet pressure was raised to its design value of 240 psia where the alternator output power average 3200 watts. Some time later the turbine exhaust pressure was raised to its design value of 7.0 psia which reduced power output to 3000 watts. The exhaust pressure was returned to 6 psia and the turbine inlet pressure raised to 265 psia where alternator output power average 3700 watts. At these conditions the turbine bearing diametral clearance started to decrease and the alternator output power started to decay. The turbine inlet pressure was therefore reduced to 240 psia. A review of turbine interstage pressures showed no changes but the boiler input power was found to be decreasing. This suggested an actual turbine flow reduction based on boiler power and alternator power but not confirmed by turbine interstage pressures.

Boiler input power and alternator output power continued to decay and eventually the first interstage pressure started to decrease indicating that the turbine first stage nozzle was plugging. Prior data suggested that an increase in superheat would reduce the plugging rate. Turbine inlet superheat was effectively increased by reducing turbine inlet pressure while maintaining the same inlet temperature. The result was a dramatic stoppage of plugging in the first stage nozzle and a settling out of vibration conditions which continued until the end of the test.

At about one third of the way through the endurance run a temporary short in a watt meter partially demagnetized the permanent magnet rotor as evidenced by a drop of 5 volts per phase. This caused a temporary increase in shaft speed vibration subsequently, and for no apparent reason the vibration dropped to a level lower than it was originally.

The bearings continued to operate very well during the 40,000 rpm run although both whirl and shaft speed vibrations were substantially greater than at 35,000 rpm. The thrust and alternator bearings remained very steady but there was a significant event regarding the turbine bearing. During the period of operation at above design turbine inlet pressure the higher rate of vapor flow through the high pressure labyrinth seal raised the shaft and journal sleeve temperatures which reduced the bearing clearance. After reducing turbine inlet pressure to ease the clearance closure problem, the estimated clearance partially recovered its original value.

Pump performance during the 40,000 rpm endurance run was not as good as it was when the unit was initially operated at 40,000 rpm prior to the 35,000 rpm endurance run. It was concluded that the jet nozzle had fouled and a post test performance test on the pump impeller confirmed this suspicion. In spite of the plugged jet nozzle the pump exceeded design requirements except for some instability at high flows.

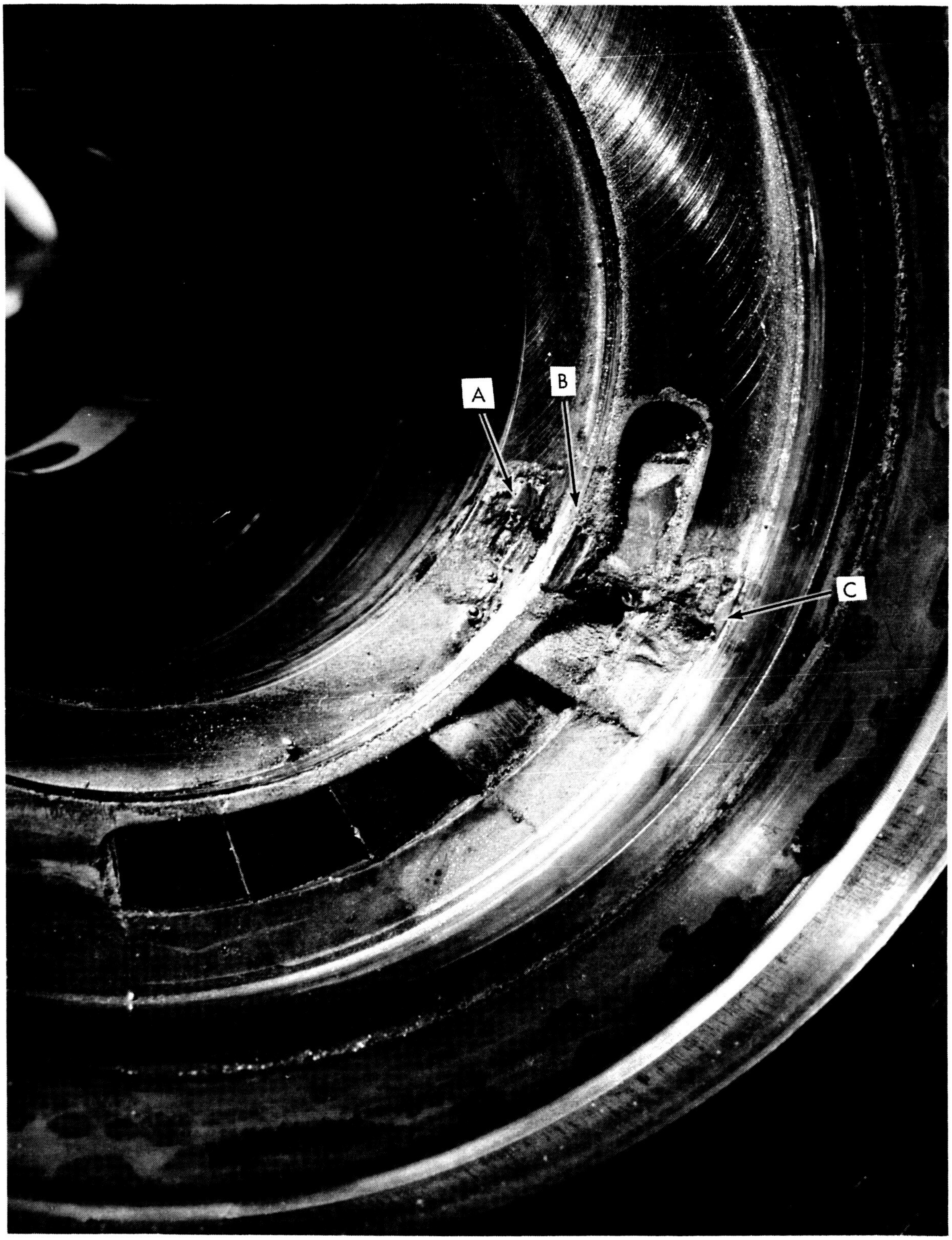
3.3.5 Shut Down

An unexpected shut down occurred at 11:13 A.M. on August 29, 1962, the first signal being a drastic change in logger temperature distributions. The turbine inlet shut-off valve failed to close on the under-speed trip-out because the long term high temperature effects had bound it in its open position. Vapor flow to the turbine was therefore closed off manually but this was not accomplished for several minutes after shaft seizure.

Just prior to the failure all control parameters were within their stipulated control ranges and the speed had been holding very steady at 40,000 rpm. Suddenly the speed dropped to 20,000 rpm, returned to 40,000 momentarily and then dropped to zero. All operating parameters reacted to the shut down as would be expected. Turbine area housing temperatures increased about 300°F and the exhaust temperature about 250°F because the shaft was no longer removing energy from the working fluid. Post test bearing calibrations showed no indications of bearing problems.

Disassembly of the unit disclosed rubbing and seizure of the first stage wheel in its wheel race. Rubbing and seizure were precipitated by a progressive accumulation of foreign metallic material on the first stage nozzle face, which subsequently interfered with the rotating first stage wheel. Page 18 is a view of the turbine inlet housing which shows the build-up of material on the nozzle. The rub marks labeled A and B registered with the blade retaining ring of the first stage wheel and the mark labeled C registered with the turbine wheel shroud ring. This shroud ring was loose from the blades and the blades and eloxed blade retaining ring were cocked on the turbine wheel hub. Although the shroud ring was loose and detached at the O.D. of the blades, the eloxed retainer ring was firmly bound on the wheel hub in the cocked position.

The bearings were free of any visible damage or wear except for a very light rub, or wipe mark, on the outboard end of the turbine bearing. Also the finish on all bearing surfaces was very good relative to the assembly finishes. The excellent condition of the thrust bearing implies that the stalling forces were largely in a radial plane. The relative lack of damage on the journal bearings indicates that the shaft was stalled in a highly concentric manner with respect to the bearing; the turbine journal for example had less than 0.001 inches diametral clearance. The condition of the



TURBINE INLET HOUSING NOZZLE DEPOSITS

FIGURE 2

bearings shows that they would not have been responsible for the major speed variations at shut down.

The pump impeller displayed cavitation erosion at the blade entrance tips, at the blade roots on the convex side, and in the face of the impeller body. The blade tip and root erosion are in areas of normally low pressure and attest to insufficient NPSH at the impeller inlet. On the other hand, the face pits may have been caused by a mismatch between actual blade curvature versus theoretical optimum curvature or by insufficient impeller NPSH. Insufficient NPSH was attributed to partial blockage of the jet boost pump.

Except for a 1/4 inch wide band of erosion on the rotor O.D. towards the pump end, the alternator sustained no damage.

There was some depositing on portions of the turbine as well as the first stage nozzle. The third stage nozzle was not affected but the first and second stage nozzle throat areas were both reduced over 20% by deposits. Metallurgical analysis of the deposits in the turbine (including those that caused the failure) revealed that the major constituent was cobalt which could only have come from the Haynes #21 housing. The balance of the system is predominatly austenitic stainless steel and corrosion was clearly evidenced in the turbine inlet housing.

The main reason for the corrosion in the turbine inlet housing was the loss of superheat which lead to mercury condensation in that area. The loss of superheat was attributed to the wrap around turbine inlet scroll which was added to more evenly distribute the heat in the turbine bearing area. It has now been demonstrated that it is no longer required and is, in fact, detrimental to the operation of the unit.

3.3.6 Conclusions

CSU I-3 test results generally fulfilled all of the stated objectives established at the outset of testing. The results demonstrated that all of the developmental improvements, with the exception of one, produced a more reliable, better performing unit than the initial CSU I-1 test unit. The total continuous 2348 hours of operation demonstrated endurance capability of better than one-quarter of the design life on the second test unit in the program. Furthermore, the post-test studies indicated no incipient failures or time effect deterioration after this exposure, where proper operating environments prevailed. Some isolated erosion and corrosion areas were evident but were related to improper operating conditions resulting from off-design testing or minor developmental design deficiencies. Material and component integrity in terms of operating stresses and dimensional stability appeared excellent after the test exposure, suggesting considerably greater endurance capability than demonstrated.

The results of the test, however, also pinpoint several areas where redesign could improve the performance and endurance capabilities of the unit. These areas are as follows:

a. Turbine inlet vapor scroll:

   The 360° turbine inlet scroll used on CSU I-3 was a redesign relative to CSU I-1 and caused excessive heat to be removed from the vapor stream before entering the first stage nozzles. This heat removal resulted in metallic deposits within the turbine which plugged nozzle throats and interfered with rotation. Return to the direct entry scroll should alleviate this problem.

b. Turbine nozzle throat areas:

   The intricate nature and miniature size of the nozzle passages combined with machining tolerances cause difficulty in fabricating to the design throat areas. In CSU I-3, the first stage nozzle throat area was about five percent small and the third stage nozzle throat area nearly 20 percent oversized. For the first stage a revision in blade height will correct the problem while in the third stage either a dimensional revision or plugging of a suitable number of passages can correct it.

c. Turbine erosion:

Erosion of the first stage labyrinth seal can be eliminated by the addition of a seal between the turbine bearing drain and vapor by-pass cavity. Third stage wheel race erosion can be eliminated by the addition of drain grooves to prevent trapping of liquid mercury. Return to the direct entry scroll and the correcting of nozzle throat areas to design values should greatly reduce turbine blade erosion.

d. Pump performance:

Erosion of the pump impeller can be reduced by a redesign of the blade profile to more nearly match the optimum blade form and also, if necessary, by substituting a harder material. Pump performance can be increased by the addition of more vanes to the back face which will reduce leakage and improve the NPSH requirement. All future units incorporate filters in the jet nozzle line to prevent fouling.

e. Screw pump seal:

Mercury leakage into the rotor gap caused drag and erosion on the rotor. A newly designed screw seal can be incorporated to eliminate this problem which is especially severe with pressurized bearing drains.

f. Turbine bearing temperature:

Temperatures in the general vicinity of the turbine bearing were excessive throughout the test. While this condition did not hurt bearing performance, it created undesirable effects such as very light corrosion of the journal sleeve and a saturated liquid vapor bearing drain condition. This problem can be alleviated by the addition of a double walled heat shield with its cavity referenced to the bearing drain environment. In addition, a seal can be incorporated to prevent entry of vapor into the bearing area and liquid into the turbine area. This could occur during operation with pressurized bearing drains and cause drag on the shaft.

g. Alternator cooling jacket:

There was a 100°F temperature gradient across the alternator housing from one end to the other. This can be flattened to a certain extent by the use of a spiral wound coil which in essence would be a counterflow heat exchanger with the coolant entering at the hot end of the alternator housing.

h. Bearing whirl:

Bearing whirl at half shaft speed persisted throughout the CSU I-3 test, both at 35,000 and 40,000 rpm. In this case no detrimental effects on bearing performance or integrity could be assigned to the whirl phenomena. However, whirl is undesirable in that it reflects displacements within the bearing which reduce the minimum loaded lube film thickness relative to a non-whirling bearing. These displacements expose the bearing to relatively greater jeopardy of rubbing, especially with any minute foreign particles that may enter the clearance. Whirl has been reduced or eliminated in Sunflower CSU's by programming assembled journalclearances to obtain operating clearances of .0010 to .0012 inches diametral. These operating clearances have been found to suppress or eliminate whirl at CSU design point operation.

i. Alternator housing flexibility:

Vibration instrumentation on the CSU I-3 test indicated greater housing sensitivity to shaft speed vibration changes by a vertically mounted accelerometer than by a horizontally mounted one. These accelerometers were mounted near the pump end of the CSU, and the CSU was mounted on its exhaust tube flange. The data suggested possible housing resonance in the vertical plane. The entire alternator, bearing support, and pump housing mass was cantilevered from the non-uniform section exhaust scroll. Bench deflection tests demonstrated that the exhaust scroll was more flexible in the vertical plane than in any other plane, and that the major delfections occurred in the exhaust scroll near the exhaust tube. These deflections can be significantly reduced by fitting the exhaust scroll with four welded struts which tie the exhaust scroll walls together and make the entire alternator housing (including the turbine nozzle section) much more rigid.

3.4 Development Redesign for CSU I-1A

CSU I-1A was a rebuild of CSU I-1 but it incorporated all the design changes described in paragraph 3.2 plus two additional changes determined from CSU I-3 testing. The two changes are:

a. The third stage nozzle throat area was adjusted to correct it to design value. This was accomplished by plugging 4 of the 28 passages.
b. A filter was added to the jet boost pump supply line to prevent jet nozzle blockage.

An additional modification was the use of ametal seal rather than the Viton A "O" ring for the static seal between the jet pump housing and the thrust bearing.

3.5 CSU I-1A Test Results and Conclusions

CSU I-1A was built entirely from original CSU I-1 parts with the exception of the bearings and the turbine inlet housing. The rebuild involved replacement of the worn bearings and rework of parts to incorporate the the changes listed in paragraphs 3.2 and 3.4. The CSU I-1A turbine inlet housing was built from a spare casting because of the economics related to inlet scroll modifications.

The purpose of the CSU I-1A component test was to gain further CSU developmental performance test data and to obtain design and off-design reference data for this unit since it was earmarked for Sunflower system testing. In general, the test objectives were as follows:

a. Perform a controlled start and attain design speed operation.
b. Conduct a power output vs. turbine inlet pressure scan at 40,000 rpm.
c. Obtain CSU pump calibrations at 40,000 rpm.
d. Conduct off-design bearing lube temperature, alternator coolant temperature, and turbine exhaust pressure tests.
e. Operate the unit with bearing drain pressures of 12 inches of mercury.

The CSU I-1A-1 test was performed in the component test rig and was started on October 3, 1962. It was voluntarily shut down on October 6, 1962 after 63 hours of continuous operation and performance testing. The unit was then removed from the compoment test rig and installed in the first Sunflower Power Conversion System (PCS I-1) where further testing was conducted in preprototype surroundings.

3.5.1 CSU I-1A-1 Component Test

All of the test objectives listed above were accomplished during the CSU I-1A-1 component test except that the initial start was more of a squirt start than a controlled start because the turbine inlet valve temporarily stuck closed. When it opened, the pressure immediately climbed to 125 psig and the speed to 37,000 rpm. It was then controlled to 20,000 rpm at 50 psig with no harm done to the unit. After reaching 40,000 rpm a power scan up to full inlet pressure of 240 psia was obtained. At this pressure the power output was 3450 watts or very nearly design value. There are two basic reasons why this unit was capable of producing this amount of power at this particular time while other almost identical units were not.

a. The turbine nozzle areas were very near to design values, especially the first stage which was 99.4 percent of design.

b. During the first half of the test (when the power was over 3400 watts) the alternator cavity drain flow was low while during the second half it was substantially higher. This was evidenced by a marked increase in the temperature of this drain flow and resulted in a reduction in power output of over 200 watts. A later section of this report will discuss the relationships between alternator cavity drain flow, its temperature, and output power.

The CSU pump was calibrated at 40,000 rpm with inlet pressures from 4.57 to 7.73 psia and, as was the case for all CSU pumps, exceeded performance specifications.

Alternator coolant temperature, average of inlet and outlet, was varied from 495° to 595°F which is 50° above and below the specified value. The stator winding temperature varied plus and minus 40° over this range and the alternator housing temperatures tended to follow the coolant temperatures. Aside from this there was no noticeable change in CSU performance.

All three bearing lube temperatures were varied individually between 350°F and 450°F. There were no appreciable changes in bearing or overall unit performance at the off-design conditions and no noticeable problems resulted.

During normal operation the turbine bearing estimated diametral clearance ran nominally at .0013 inches. There were fluctuations in flow and pressure but this was apparently a rig control problem as the relationship between them didn't change. The alternator bearing clearance started at .0009 inches but during most of the test ran between .00095 to .0010 in. The clearance of both journal be bearings increased as power was increased at the start of the test while the thrust bearing was relatively unaffected by this or any other parametric change.

All three bearings were simultaneously flooded by imposing a 12 inch liquid head of mercury on the bearing drain line. The clearances of both journal bearings appeared to increase appreciably during flooding but the thrust bearing performance didn't change. There was a 400 watt loss in CSU power output that was completely recovered when the unit was returned to normal operation.

Turbine exhaust pressure was varied between 5.2 and 7.5 psia and this resulted in an output power variation of 300 watts.

Throughout the test the vibration levels were quite low (less than 1 "g"); however, the half speed "g" level was almost as high as the shaft speed "g" level. This did not appear to have any detrimental effect on bearing performance or on overall CSU performance. Post test bearing calibrations agreed very well with pre-test checks.

3.5.2 PCS I-1 System Test

The reference data obtained during the CSU I-1A-1 test indicated that CSU I-1A would be capable of operating under certain off-design conditions imposed by the PCS I-1 pre-prototype power conversion system. This conclusion was verified during the 47 hours of system testing which began on November 27, 1962 and was concluded on December 19, 1962. TRW ER-5396, Revision A, "Sunflower Power Conversion Topical", describes this test in detail.

PCS I-1 System test consisted of three separate runs identified as PCS I-1-1, PCS I-1-2, and PCS I-1-3 which are summarized as follows:

a. PCS I-1-1

   This initial run lasted a total of 3 hours and consisted of 2 separate starts. During this run the power output reached 2000 watts at 1478 cps and 206 psig turbine inlet pressure. The test was terminated when the condenser no longer could control exhaust pressure and turbine exhaust temperatures reached 710°F. The second run was terminated shortly after it started because of a valve problem.

b. PCS I-1-2

   This run consisted of 10 separate starts with actual running time totaling 44 hours. The CSU operated at various speeds up to 40,000 rpm and usually had to operate at off-design turbine exhaust pressures and with the bearing drains pressurized. Its performance was not as good as it was during the component test because of shaft drag due to flooding caused by pressurized bearing drain lines. The test had to be shut down and restarted ten times due to condenser performance as effected by other system parameters. The result was an increasing turbine exhaust pressure which climbed to an unsafe level. A shut down and restart was required to regain satisfactory system operation.

c. PCS I-1-3

Prior to this test certain test rig modifications were made to aid in evaluating system performance. This run consisted of only one start and was very short in duration. The CSU spun up to 30,000 rpm momentarily, stopped and could not be restarted. Bearing calibrations indicated that the estimated diametral clearances of the bearings were still the same as prior to the CSU I-1A-1 component test. An analysis of a mercury sample from the system disclosed the presence of lithium which indicated a boiler failure. The CSU was removed from the system for inspection.

Lithium-mercury amalgam was found in the CSU but did not cause damage to anything but the glass bore seal of the alternator stator. It severly attacked the glass and caused interference between the rotor and stator which prevented shaft rotation. After the component parts were cleaned their condition was excellent with the exception of the stator. There was no measurable bearing wear and erosion was nonexistent. The unit was reassembled with all the original parts except for a new alternator stator.

In summary, the results of CSU I-1A testing indicate the ability of the unit to fulfill its performance requirements provided certain previously identified design improvements are incorporated. No problems were associated with the turboalternator tests except to lower power output when coupled with system components. This discrepency should be eliminated by the incorporation of the design improvements discussed in paragraph 3.3.6. With regard to other system requirements, even without the recommended design improvements, CSU I-1A demonstrated the ability of the unit to withstand the equivalent of ten squirt starts and a total of 13 starts with no harmful effects.

3 6 Developmental Redesign for CSU I-3A

There were no changes of any significance in CSU I-3A as compared to CSU I-1A. The unit is described in paragraph 1.3 which means that it had incorporated the design improvements described in paragraphs 3.2 and 3.4.

4.0 CSU I-3A TEST RESULTS

Testing of CSU I-3A was primarily endurance running with most parameters constant at design conditions. There were three separate runs which totaled 4325 hours and each of the runs will be discussed regarding operation of the unit and significant events which occurred. The test history is presented graphically with supporting discussion to cover these events as well as operational performance of the CSU at various stages during the test.

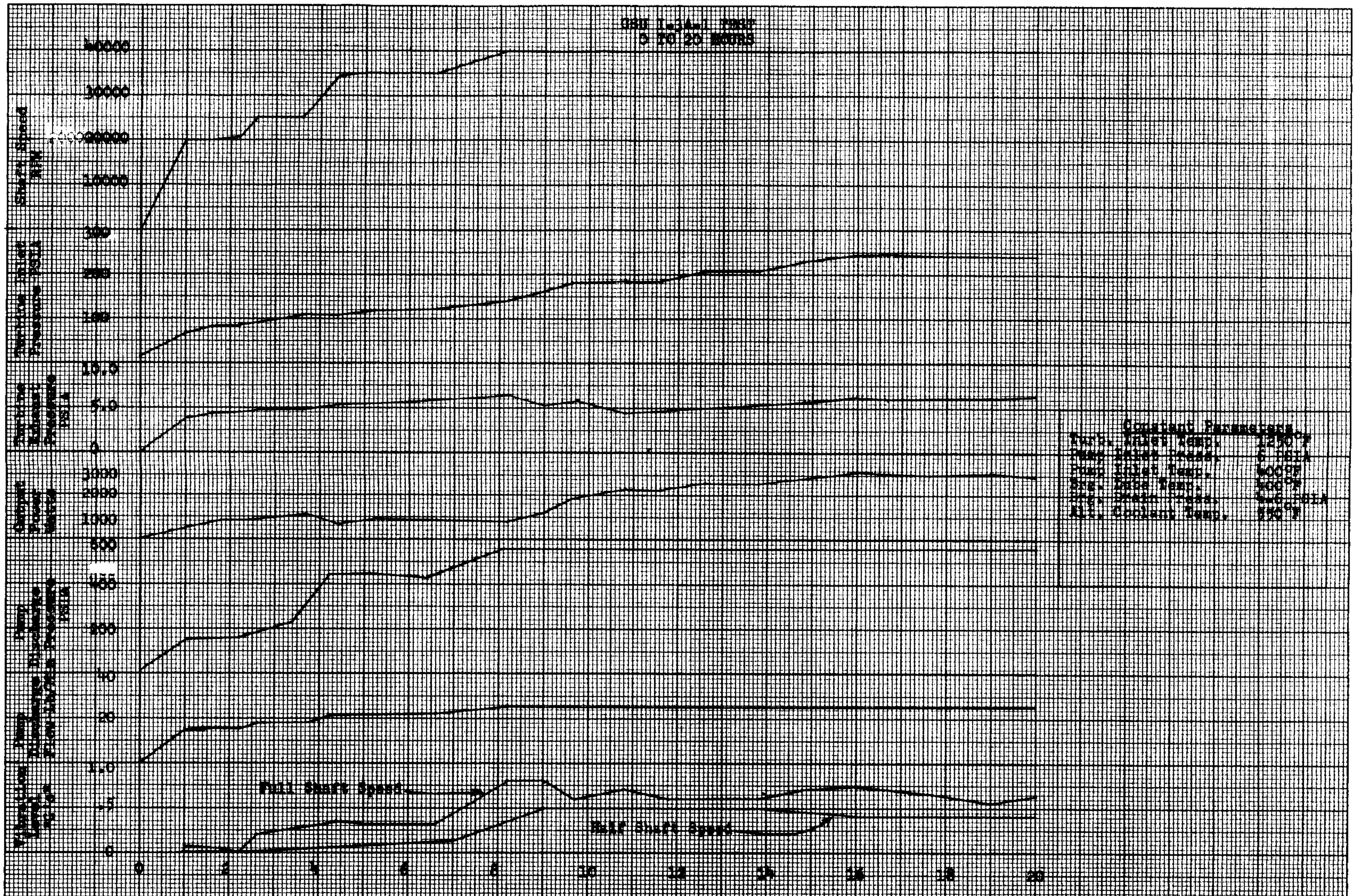
The graphs are time plots of significant variables with less significant or relatively constant parameters tabulated in the margins. Although the plots are averages and appear as straight lines, they are generally accurate representations of operating conditions at any point in time, relfecting the degree of constancy of unit operation and test rig control.

4.1 CSU I-3A-1 Test

CSU I-3A-1 test was started on February 5, 1963 and continued for 769 hours when it was voluntarily shut down because of apparent blockage of of the first stage turbine nozzle. Figures 3, 4, and 5 are time plots of the various portions of the test run which will be discussed in the paragraphs below.

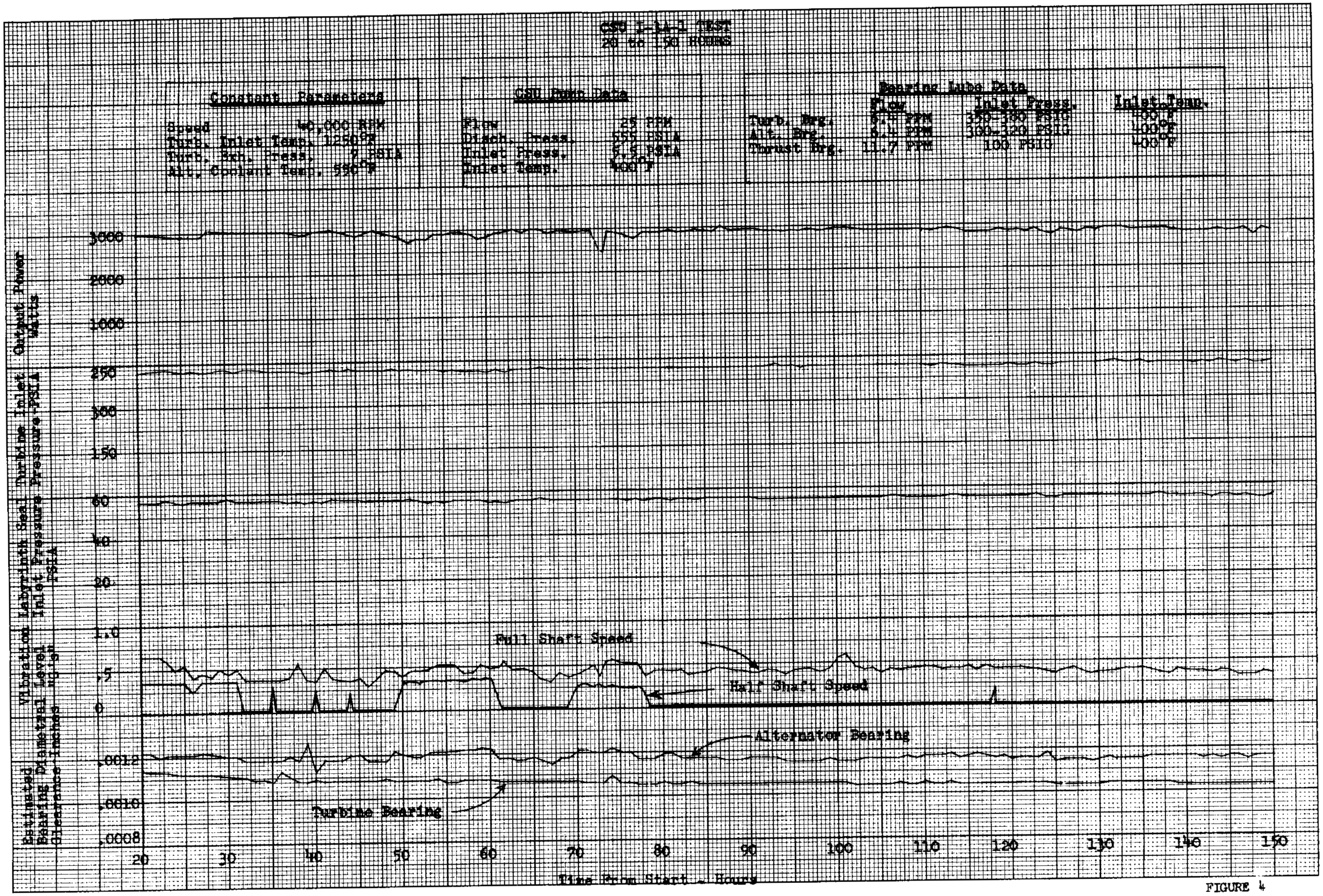
4.1.1 Start-Up and Speed Climb (0 to 20 hours)

Figure 3 is a plot of the first 20 hours of operation showing speed and power increases as well as vibration and pump data. The start transient itself (not shown in the plot) was very smooth and easily controllable with turbine inlet pressure and simulated load resistance. The first speed plateau was 20,000 rpm with 500 watts output power at 60 psig turbine inlet pressure. It was noted that the speed control was not operating but the reason was not apparent at the time. The next plateau was 25,000 rpm with 98 psig inlet pressure and 1080 watts output without the speed control. Since the



CSU 1-3A-1 TEST
0 TO 20 HOURS
Shaft Speed RPM
Turbine Inlet Pressure PSIA
Turbine Exhaust Pressure PSIA
Output Power Watts
Pump Discharge Pressure PSIA
Pump Discharge Flow Lb/Min
Vibration Level
Full Shaft Speed
Half Shaft Speed
Constant Parameters
Turb. Inlet Temp. 1250°F
Pump Inlet Press. 6 PSIA
Time From Start - Hours

FIGURE 3



CSU I-1A-1 TEST
20 to 150 HOURS
Constant Parameters
Speed 40,000 RPM
Turb. Inlet Temp. 1250°F
Turb. Exh. Press. 9 PSIA
Alt. Coolant Temp. 550°F
CSU Pump Data
Flow 25 PPM
Disch. Press. 555 PSIA
Inlet Press. 5.5 PSIA
Inlet Temp. 400°F
Bearing Lube Data
Flow
Inlet Press.
Inlet Temp.
Turb. Brg. 6.4 PPM 350-380 PSIG 400°F
Alt. Brg. 6.4 PPM 300-320 PSIG 400°F
Thrust Brg. 11.7 PPM 100 PSIG 400°F
Output Power Watts
Turbine Inlet Pressure PSIA
Labyrinth Seal Inlet Pressure PSIA
Vibration Level "g"
Estimated Bearing Diametral Clearance Inches
3000
2000
1000
250
300
150
60
40
20
1.0
.5
0
.0012
.0010
.0008
Full Shaft Speed
Half Shaft Speed
Alternator Bearing
Turbine Bearing
20
30
40
50
60
70
80
90
100
110
120
130
140
150
Time From Start - Hours

FIGURE 4

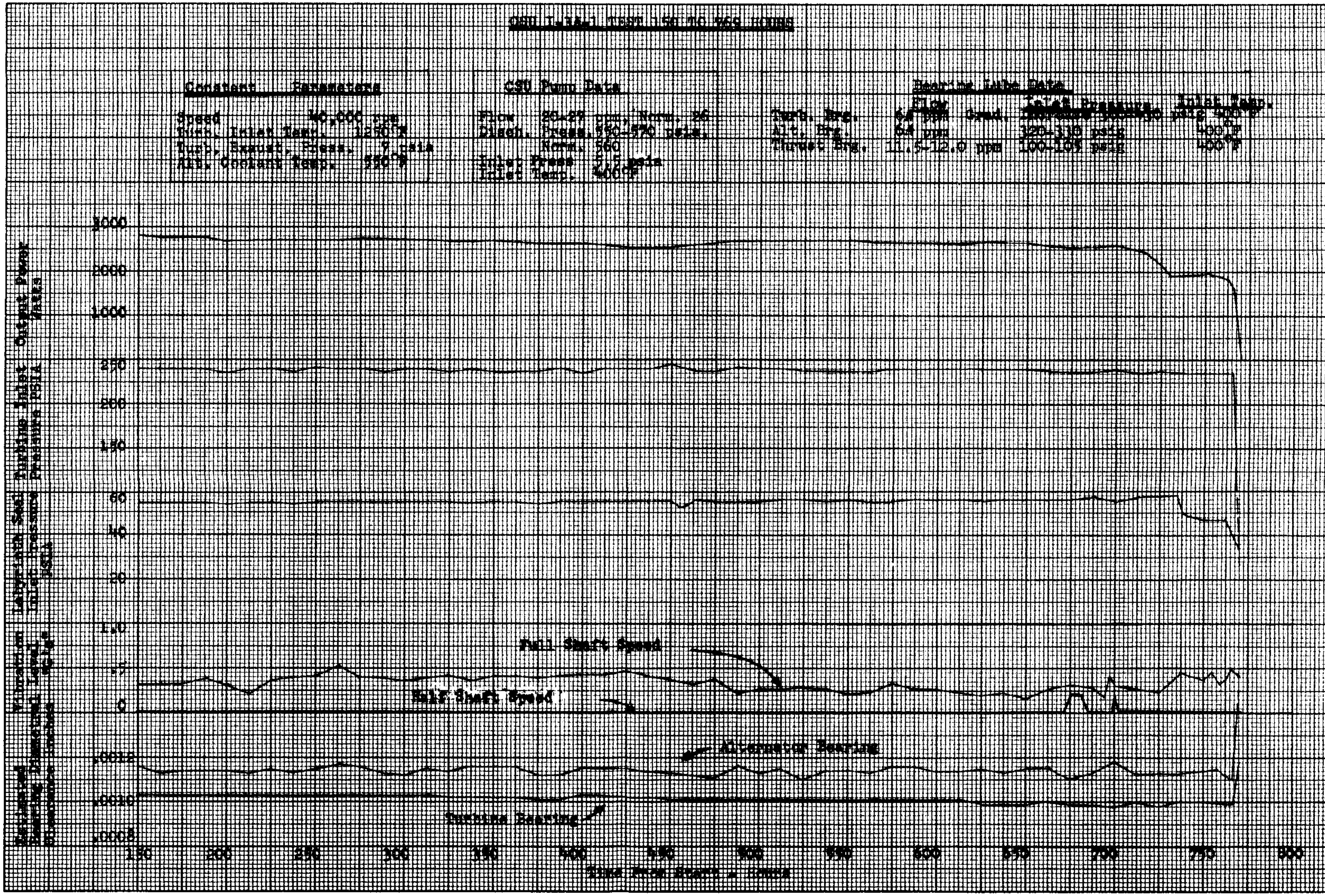

Constant Parameters
Speed 40,000 rpm
Turb. Inlet Temp. 1250°F
Turb. Exhaust Press. 9 psia
Alt. Coolant Temp. 550°F
CSU Pump Data
Flow 20-29 ppm, Norm. 26
Disch. Press. 550-570 psia, Norm. 560
Inlet Temp. 400°F
Bearing Lube Data
Flow
Inlet Pressure
Inlet Temp.
Turb. Brg.
Alt. Brg. 64 ppm 320-330 psig 400°F
Thrust Brg. 11.5-12.0 ppm 100-105 psig 400°F
Output Power Watts
Turbine Inlet Pressure PSIA
Labyrinth Seal Inlet Pressure PSIA
Vibration Level G's
Estimated Bearing Diametral Clearance Inches
3000
2000
1000
250
200
150
60
40
20
1.0
.5
0
.0012
.0010
.0008
Full Shaft Speed
Half Shaft Speed
Alternator Bearing
Turbine Bearing
150
200
250
300
350
400
450
500
550
600
650
700
750
800
Time From Start - Hours

FIGURE 5

first mode criterical is approximately 30,000 rpm the speed was increased to 35,000 rpm at a power level of 1000 watts with 105 psig turbine inlet pressure. At this time the CSU pump was coupled; i.e., the pump supplied its own jet boost instead of receiving it from the test rig. The speed control was still not operating properly and appeared to be causing a voltage unbalance between phases.

After an elapsed time of 8 hours the unit was operating at 40,000 rpm with 1000 watts output at a turbine inlet pressure of 125 psig. The turbine inlet pressure was gradually increased until, after 16 hours of elapsed time, it was at the design value of 240 psia. At this pressure and a turbine exhaust pressure of 6 psia the output power was 3100 watts.

During the first 20 hours the unit performed quite well in spite of the problem with the speed control. Output power was below design but this was mainly due to any undersized first stage turbine nozzle (91% of design area). Pump flow was not raised to its design value of 34 ppm but at a flow of 25 ppm and discharge pressure was 560 psia vs. a 450 psia design value.

Bearing performance was very satisfactory with the turbine bearing diametral clearance nominally .00115 inches and the alternator bearing .0012 inches. Half speed whirl and shaft speed "g" level were the same but their magnitude of .4 and .5 "g's" was relatively low. No attempt was made during this time period to suppress whirl by increasing the bearing flow.

4.1.2 Parametric Testing (20 to 150 hours)

The time period from 20 to 150 hours was devoted mainly to parametric testing aimed at fulfilling the test objectives. All of the objectives aside from squirt start and endurance testing were accomplished during this period. The plot on Figure 4 showing unit operation for this time period does not show the parametric deviations but does show the average performance of the turboalternator unit.

Generally, the unit was operating at design conditions and when off design conditions were imposed they were maintained only long enough to sufficiently evaluate the effect on overall performance. Shortly after the beginning of this time period the speed control was removed for examination and found to have an improperly connected neutral line. After this line was moved to its correct position the control was reinstalled and performed very well for the balance of the testing of CSU I-3A.

Each of the parametric tests will be discussed individually in the following sections with regard to the amount of variation from design and the effect on unit operation. Also a subsequent section describes other significant events which occurred during this period.

a. Pump Calibration

Approximately 21 hours after startup the CSU pump was calibrated at three different inlet pressure; 3.5, 4.7, and 5.7 psia. At 3.5 psia inlet pressure the flow was varied from 33.6 ppm where the pressure was 475 psia to 25.2 ppm where the pressure was 550 psia. With an inlet pressure of 4.7 the flow range was 25 to 38 ppm with discharge pressures of 555 and 500 psia respectively. At an inlet pressure of 5.7 psia the flow ranged from 43 ppm at 475 psia discharge to 20 ppm at 575 psia. The pump operation was smooth over this operating range and its performance equaled or exceeded design requirements especially at the higher inlet pressures.

b. Bearings

Aside from the off-design parametric tests, operation of all three bearings was steady during this time period. Lube low for both journal bearings was constant at 6.4 ppm with the thrust bearing flow holding at 11.7 ppm once the unit reached full power. Beyond this point there were no more changes in thrust bearing flow or pressure. The alternator bearing supply pressure gradually rose from 300 to 320 psig which indicates a decrease from .0012 to .00115 inches diametral clearance. Turbine bearing inlet pressure also gradually increased from 350 to 380 indicating a reduction in diametral clearance from .00115 to .00105 inches. These reductions in journal bearing diametral clearances did not cause concern since past experience had shown that this period of time was required for the clearance to stablize.

Several parametric tests concerning the bearings were performed as specified in the test objectives, the results of which follow. Just prior to these tests half speed bearing whirl decreased to zero which resulted in a slight increase in lube supply pressure to both journal bearings.

1. Turbine Bearing Lube Flow Variation
   Turbine bearing lube flow was varied over the range of 5 to 8 ppm. When it was raised from 6.4 to 8 ppm the diametral clearance increased slightly (about 4%) but there was no other effect on CSU operation. Decreasing the flow from 6.4 to 5 ppm did not decrease the diametral clearance but half speed whirl reappeared. There was no other change in CSU operation.
2. Alternator Bearing Lube Flow Variation
   As in the case of the turbine bearing the alternator bearing lube flow was varied from 5 to 8 ppm. In this case however, the diametral clearance decreased slightly (about 5%) at 5 ppm as well as increasing slightly (about 8%) at 8 ppm. There were some supply pressure fluctuations at 5 ppm. Aside from this and the reappearance of whirl at the low flow there was no other change in unit operation.
3. Thrust Bearing Lube Flow Variation
   The thrust bearing flow was not varied because test rig limitations would not permit raising the flow above its normal 12 ppm and it could not be lowered because it was operating at its arbitrary low inlet pressure limit of 100 psig.
4. Bearing Lube Temperature Variation

Lube inlet temperatures of all three bearings were simultaneously reduced from their normal 400°F level to 375 and to 350°F. This resulted in a very slight increase in the alternator bearing diametral clearance and a 40 psi increase (to 140 psig) in thrust bearing supply pressure. There was no effect on the turbine bearing. Except for localized temperature changes caused by the lube temperature variation, there was only one significant change in performance. This was a 100 watt decrease in alternator output power at 350°F lube temperature probably the result of increased flow into the alternator cavity causing shaft drag. It did however, return to normal when the lube temperature was brought back to 400°F.

The lube temperatures were then raised (again simultaneously) to 425 and then 450°F which produced results opposite to those at the reduced temperature but to a lesser degree. There was a very slight reduction in alternator bearing clearance and a 15 psi decrease in thrust bearing supply pressure. The turbine bearing was somewhat unsteady during the transition to 450°F but once the temperature stablized its operating characteristics were unchanged. There were no changes in other operating conditions or parameters aside from some localized temperature changes caused by the lube temperature variation. The bearing drain lines were plumbed such that a drain pressure of up to 12 inches of mercury could be imposed on the bearings individually or simultaneously. The full 12 inches was applied to the alternator-thrust bearing drain line and this flooding of the bearing caused liquid to pass through a dynamic seal resulting in a drag on the alternator rotor and a loss of 500 watts power output. In addition to the power loss there was a slight decrease in alternator bearing flow and minor changes in some of the housing temperatures. The 12 inch head was then imposed on the turbine bearing drain thus simultaneously flooding all three bearings. This resulted in an additional 50 to 70 watt loss in power output. Under these conditions the alternator cavity drain flow was measured at 3.75 to 4.0 ppm which explains the loss in power. These results reaffirm the necessity of having effective seals at both the turbine and alternator bearings.

Bearing drain pressure was gradually reduced to its normal value and operating conditions returned to their original values. It took two or three hours for some parameters such as the turbine interstage pressures to return to normal; and half speed whirl, which had reappeared, didn't diminish for several hours. All of the power lost during flooding of the bearings was regained and in fact was eventually somewhat higher when the alternator cavity drain flow decreased below its pre-flooded bearings level.

c. Alternator Coolant Temperature Variation

Alternator coolant inlet temperature was varied over the range of 450 to 580°F, its normal operating level being 540°F. At the two extremes the outlet temperature was 510 and 590°F. This excursion didn't appear to affect the unit except for a corresponding decrease and increase in alternator housing temperatures.

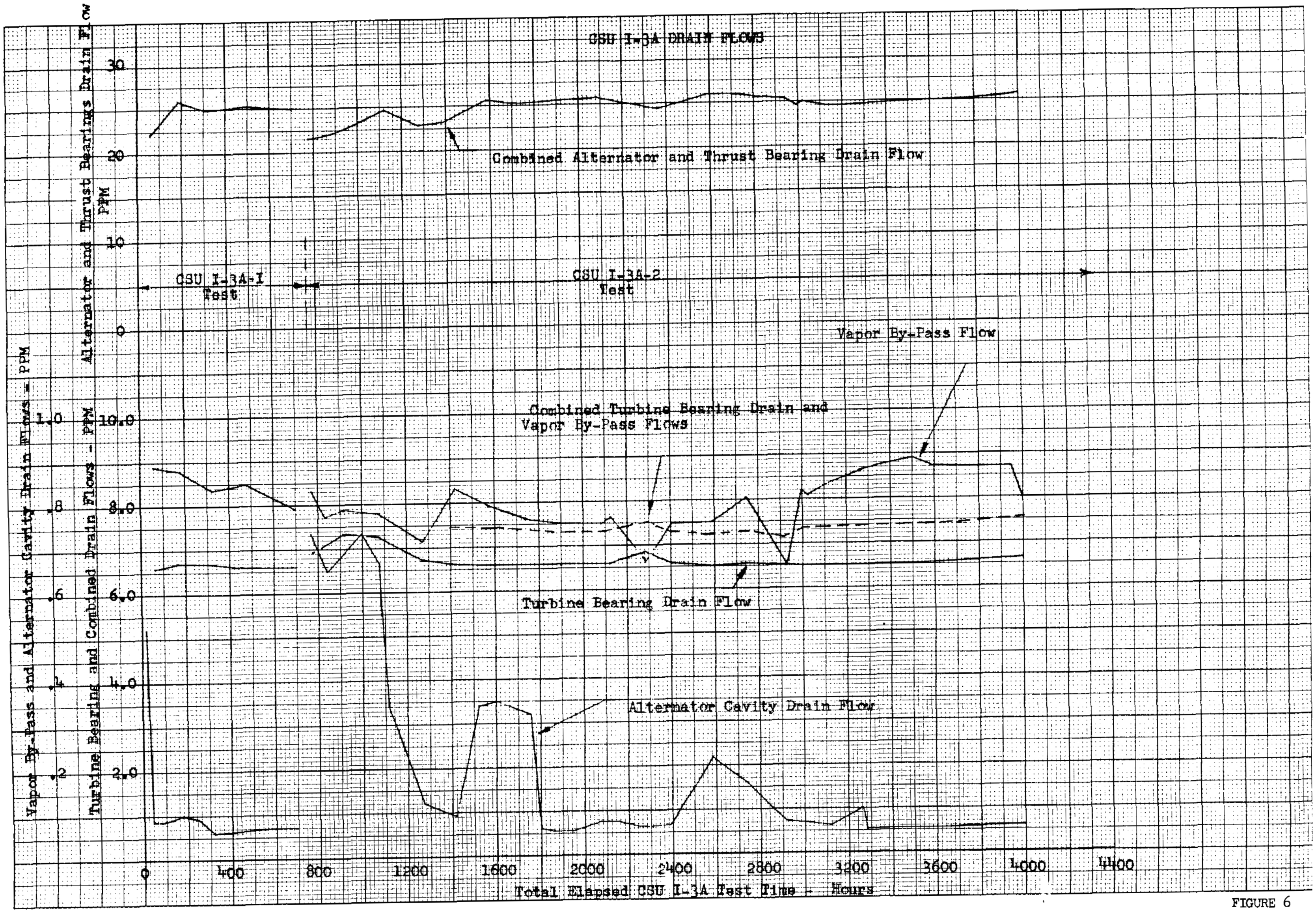
d. Turbine Exhaust Pressure Variation

The turbine exhaust pressure was reduced from its usual level of 7 psia to 5.5 psia which lowered the exhaust temperature from 605 to 585°F. This resulted in a gain of 200 watts in output power. Other parameters affected were the bearing drain and vapor cavity pressures which dropped slightly as did the turbine exhaust scroll temperatures. When the exhaust pressure was returned to normal they immediately returned to their original values.

e. Drain Flow Measurements

CSU I-3A was the first unit for which provisions were made to measure the various drain flow; i.e., alternator cavity drain, turbine vapor by-pass drain, turbine bearing drain, and the combined alternator and thrust bearing drain. In previous units the alternator cavity drain flow and turbine vapor by-pass flow both discharged into the turbine exhaust and their magnitudes were unknown. These and the bearing drains of CSU I-3A were plumbed such that they normally flowed back into the test rig but, when desired, could be collected individually and weighed for a flow measurement. There were several benefits derived from having this drain flow measurement capability. During previous tests there were occasions when flow measuring venturis changed calibration and in the case of bearing flow this could give misleading information concerning bearing operation. Periodic measurements of the bearing drain flows during the CSU I-3A test double checked the readings of the bearing flow venturis. Alternator cavity and turbine vapor cavity drain flow measurements provided an insight into the effectiveness of the dynamic seals within turboalternator unit. Also, in the case of the alternator cavity drain flow, it was especially important because the magnitude of this flow had a significant effect on CSU power output.

Throughout the testing of CSU I-3A, periodic measurements of all drain flows were made and the results are presented graphically in Figure 6. During the first 150 hours the turbine bearing drain flow nominally was 6.6 ppm which agreed very well with the indicated 6.4 ppm lube flow. This slight discrepancy remained constant for all CSU I-3A testing. The alternator and thrust bearing drain flow was somewhat erratic for the first 200 hours but settled out at 25 ppm which was 7 ppm more than the total alternator and thrust bearing lube flow. This discrepancy remained constant throughout the CSU I-3A test indicating that this drain flow was a valid check on the bearing lube flow accuracy. Backface leakage from the CSU pump accounted for the additional 7 ppm flow. Vapor by-pass flow was approximately 0.85 ppm. The alternator cavity drain flow, however, was quite erratic and as previously mentioned had an appreciable effect on CSU performance. Initial measurements of this flow indicated



CSU I-3A DRAIN FLOWS
Combined Alternator and Thrust Bearing Drain Flow
CSU I-3A-I Test
CSU I-3A-2 Test
Vapor By-Pass Flow
Combined Turbine Bearing Drain and Vapor By-Pass Flows
Turbine Bearing Drain Flow
Alternator Cavity Drain Flow
Vapor By-Pass and Alternator Cavity Drain Flows - PPM
Turbine Bearing and Combined Drain Flows - PPM
Alternator and Thrust Bearings Drain Flow PPM
30
20
10
0
10.0
8.0
6.0
4.0
2.0
1.0
.8
.6
.4
.2
0
400
800
1200
1600
2000
2400
2800
3200
3600
4000
4400
Total Elapsed CSU I-3A Test Time - Hours

FIGURE 6

a flow of .5 ppm but after a short period of time settled out slightly less than .1 ppm. In the sections describing later portions of the test the alternator cavity drain flow will be discussed in more detail.

f. General Comments

The preceeding sections which discussed the various parametric tests in general described most of the testing which took place during the first 150 hours. However, there were a few additional occurances of interest which took place. One of these was the determination of which direction the thrust bearing is loaded. The way chosen to do this was to raise the alternator bearing drain pressure thus loading the shaft in the direction of the turbine. This was accomplished by referencing the drain pressure, which had been running 20.5 in. Hg vacuum, to turbine exhaust pressure which had been running about 17.5 in. Hg vacuum. When this was accomplished the thrust bearing pressure rose slightly at constant flow indicating a shift towards a more centered position. Therefore the thrust bearing is normally loaded towards the pump end of the CSU. Two additional things were noted during the time the drain pressure was raised. One was an increase in seal leakage as evidenced by a loss in power and increased alternator cavity drain flow. The other was the elimination of half speed whirl which confirms that whirl was motivated by the alternator bearing.

Another phenomenon noted shortly after the thrust bearing investigation was the effect of CSU pump flow on alternator cavity drain temperature. The flow was varied between 26 and 35 ppm and it was expected that at the higher flow, and therefore lower pressure, the leakage into the alternator cavity would be reduced. However, it had the opposite effect. Apparently it caused the alternator bearing drain pressure to increase resulting in greater flow through the dynamic seal into the alternator cavity.

The periodic checks of alternator cavity drain flow indicated a direct relationship between this flow and its temperatute. As previously mentioned high cavity drain flow appreciably affected CSU power output. Since the alternator bearing drain pressure greatly influenced the amount of alternator cavity drain flow it became apparent that to keep the cavity drain flow at low level the alternator bearing drain pressure would have to be controlled. Fortunately this pressure could be controlled in the following manner. As described above, opening the alternator bearing drain "degas" valve, which referenced the bearing drain pressure to condenser pressure, would raise the drain pressure. Also, as described above, lowering the pump flow lowered the drain pressure. Therefore, the pressure could be raised or lowered as required and after a period of experimentation it was found that maintaining it in the range of 20 to 22 in. Hg vacuum kept the alternator cavity drain flow at an acceptable level. It was not desirable to allow the pressure to go below the low limit because of the possibility of boiling within the drain line.

4.1.3 Endurance Run (150 to 769 hours)

Having completed the parametric testing the next objective was to determine the endurance capability of the unit. Thus the balance of the CSU I-3A-1 test was devoted to steady state operation or at near design conditions. Maintaining control parameters within their specified range was accomplished with relative ease. Fluctuations of incoming plant electrical power was the main cause of variation from the nominal control levels.

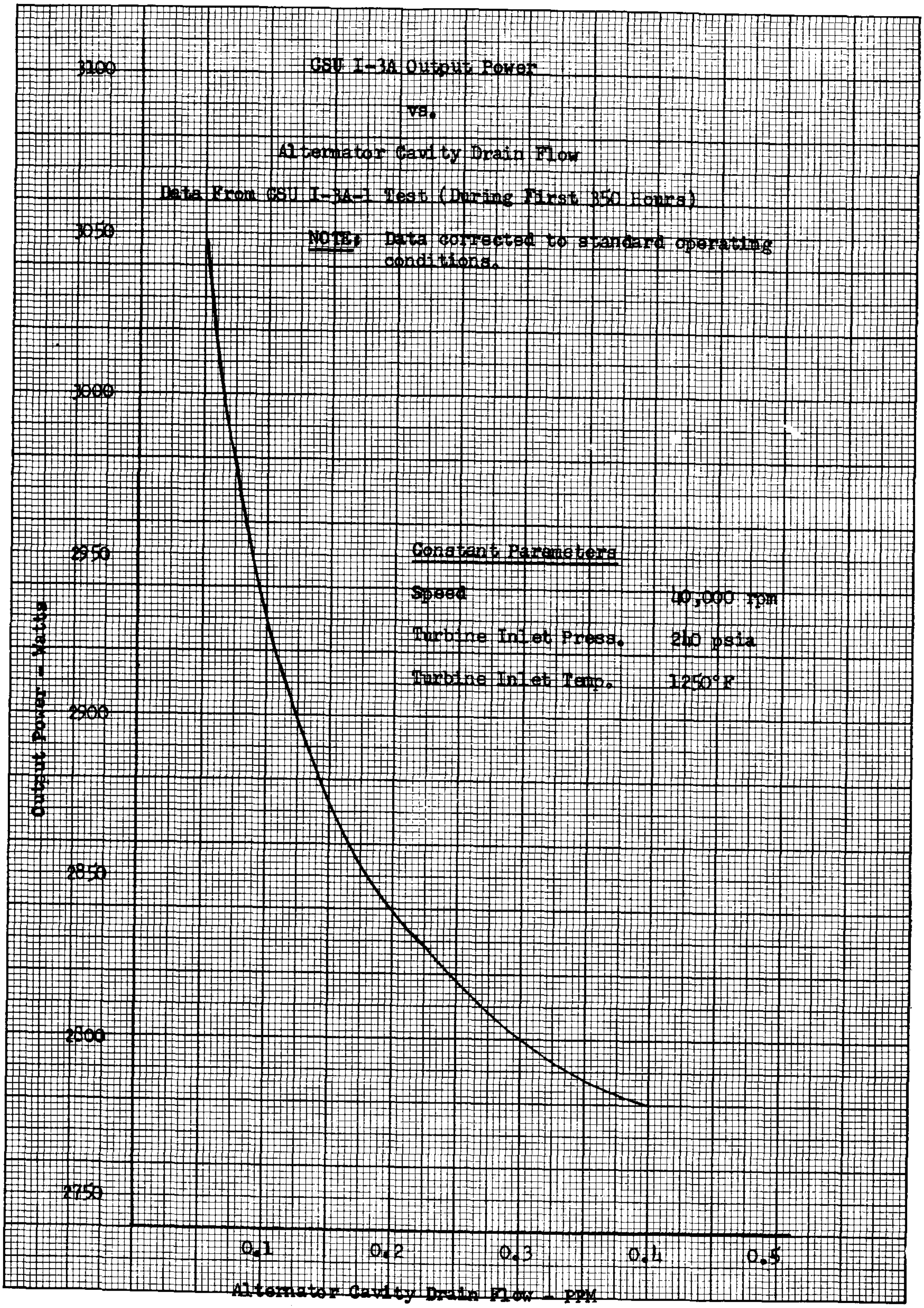
Turbine flow was controlled by adjusting the power input into the boiler which determined the amount of mercury vapor generated. Bearing lube temperatures, alternator coolant temperature, and CSU pump inlet temperature were controlled by individual heaters. Bearing and pump flows were controlled by metering valves. Turbine exhaust pressure was controlled through the use of a vacuum pump. As explained in the section 4.1.2 it was found that in order to keep the alternator cavity drain flow at a low level the alternator bearing drain pressure had to be held within certain limits. This could be accomplished by cracking the alternator bearing drain "degas" valve to increase this drain pressure or by lowering pump flow. Occasionally it was difficult to obtain an equilibrium point which held the pressure in its 20 to 22 in. Hg vacuum range. The interactions between the alternator bearing drain pressure, alternator cavity drain flow and CSU output power are shown in Figures 7 and 8. However, in general only trim adjustments were required to compensate for the plant power fluctuations.

Figure 5 shows the plots of significant parameters versus time during the endurance run. A gradual decline in output power can be noted. It was suspected that the heat loss from the turbine inlet housing was sufficient to result in wet vapor entering the first stage nozzle. This condition can result in corrosion products collecting in, and partially blocking the nozzle passages. If true, the reduced nozzle throat area would lower the weight flow through the turbine and thus reduce its output. The turbine inlet housing heater was activated to alleviate this problem.

Throughout the test the turbine bearing diametral clearance exhibited an apparent decrease. The magnitude was extremely small and did not pose any operational problem other than to arouse suspicion. As a result it was carefully monitored.

To more accurately evaluate the performance of the unit at a given instant, measurements of the variation in output power as a function of turbine inlet and exhaust pressures were made. Using this information the power output could be corrected to standard operating conditions for comparison with previous data. Applying these corrections and taking into account the effect of alternator cavity drain flow showed that after 550 hours the decay in power output was 300 watts. However, it had remained constant for the preceeding 300 hours which coincided with the reactivation of the turbine housing heater.

At an elapsed time of 725 hours, however, there was a stepchange



CSU I-3A Output Power
vs.
Alternator Cavity Drain Flow
Data From CSU I-3A-1 Test (During First 350 Hours)
NOTE: Data corrected to standard operating conditions.
Constant Parameters
Speed 40,000 rpm
Turbine Inlet Press. 240 psia
Turbine Inlet Temp. 1250°F
Output Power - Watts
3100
3050
3000
2950
2900
2850
2800
2750
0.1
0.2
0.3
0.4
0.5
Alternator Cavity Drain Flow - PPM

FIGURE 7

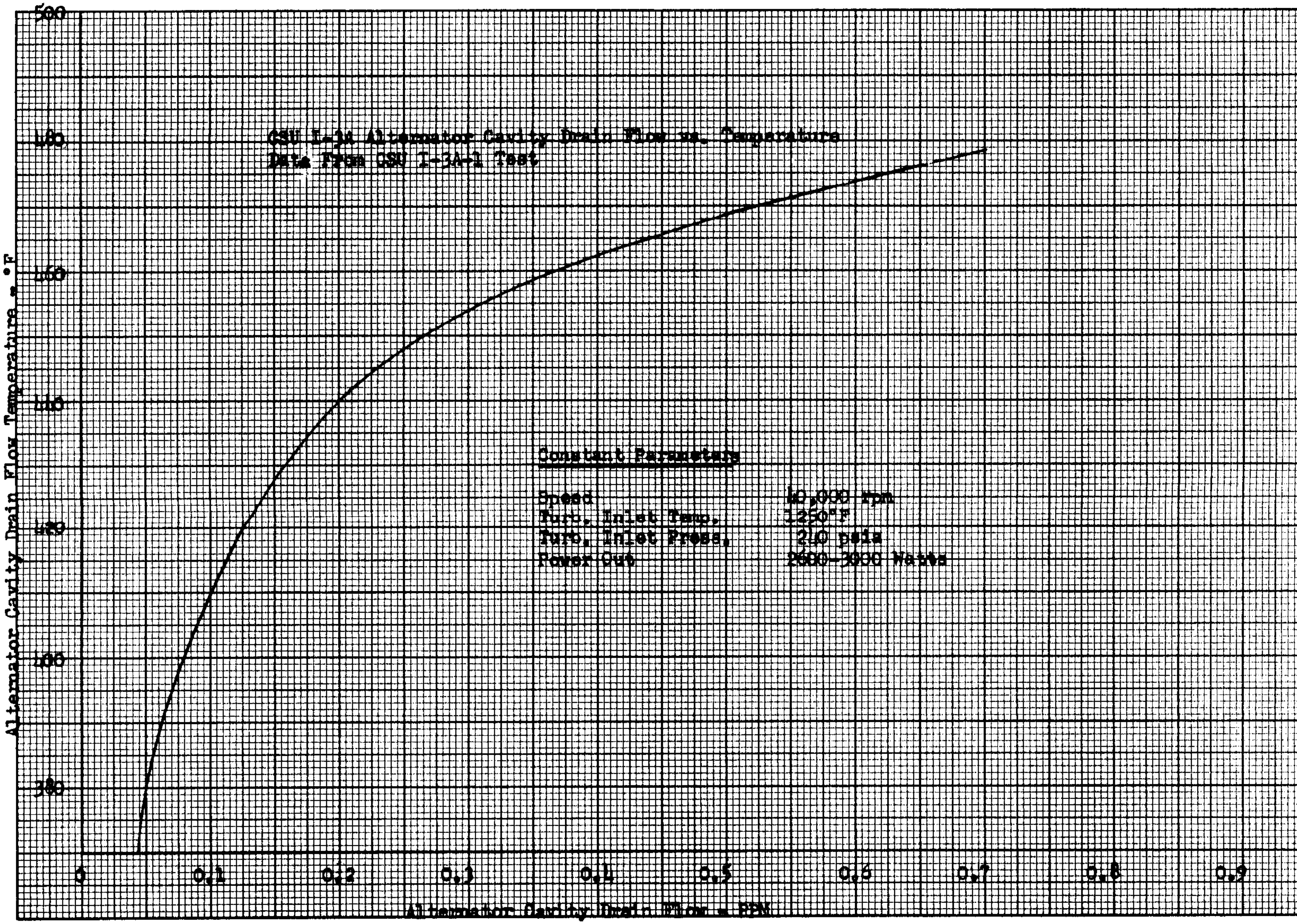

CSU I-3A Alternator Cavity Drain Flow vs. Temperature
Data From CSU I-3A-1 Test
Constant Parameters
Speed 40,000 rpm
Turb. Inlet Temp. 1250°F
Turb. Inlet Press. 240 psia
Power Out 2600-3000 Watts
500
480
460
440
420
400
380
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
Alternator Cavity Drain Flow Temperature - °F
Alternator Cavity Drain Flow - PPM

FIGURE 8

in power of about 600 watts. At the same time the turbine interstage pressures and boiler power both dropped off which indicated reduced vapor flow through the turbine. Approximately 40 hours later there were additional step reductions in power and at 769 hours the power had dropped to 1600 watts. All indications were that the first stage nozzle was plugging. To avoid operation at low power and to preclude the possiblility of shaft seizure the test was shut down. The unit was removed from the test rig and partially disassembled.

4 1.4 Disassembly

Aside from the step changes in power the CSU was performing very satisfactorily right up until shut down. It was strongly suspected that the first stage nozzle was partially blocked and this suspicion was fonfirmed when the turbine inlet housing was removed. Discrete particles, ranging in size from .040 to .060 inches, were found lodged in the nozzle passages. These particles were removed as was a black film found on the turbine bearing journal. This film had no measurable thickness and was easily polished off. It is likely that this film was responsible for the apparent decrease in turbine bearing diametral clearance.

Prior to reassembly of the unit, the first stage nozzle throat area was measured and found to be 87.5% of design as compared to 91% originally. Apparently some corrosion products deposition had occurred prior to actuation of the turbine housing heater which subsequently could not be removed by the normal cleaning procedure. This observation also explains the early (300 watt) drop in power. This unit was reassembled with all original component parts and returned to the test rig for continuation of the endurance run.

4.2 CSU I-3A-2 Test

CSU I-3A-2 test was a continuation of the endurance testing of CSU I-3A, the objective being to accoumulate a minumum of one years operation. The test was involuntarily shut down, however, after 3556 hours of continuous operation because of a municipal power failure. At the time of the shut down the unit had accumulated 4325 hours or the equivalent of 6 months operating time. Although the operation of the unit had been compromised during the final 200 hours of the test there were no indications of incipient failure and it appeared to be capable of continuing the endurance run for an extended period of time.

As in the case of the CSU I-3A-1 test the test history is presented graphically and significant events will be discussed along with operational performance of the CSU at various stages during the test.

4.2.1 Test Rig Modifications

Prior to reinstallation of the unit into the test rig certain rig modifications were made to prevent the formation of the particles which were responsible for the shut down of the CSU I-3A-1 test. A review of the test rig data indicated that a rig shut down prior to that test resulted in a back flow of fluid from the boiler inlet up through the centrifugal separator and the discharge of the superheater and into the area of the metering and shut-off valves at the CSU. The deposits found in front of the shut-off valve appear to have been formed by floating of corrosion products on the mercury surface during this back flush and subsequent evaporation of the mercury from the deposits. With time these products were eroded or dislodged by the vapor flow and subsequently progressed into the turbine housing causing the step changes in power. As a result of the formation of particles, several changes in the concepts of the test rig were incorporated to prohibit the formation of products on the metering and shut-off valves and to eliminate the possibility of an invertent back flow to the turbine area. These changes were as follows:

a. Removal of metering and shut-off valves from the position immediately upstream of the turbine to a location at the exit of the boiler prior to the superheater which lowered their operating temperature.

b. Placement of the by-pass function of the test rig in a similar location.

c. The use of a shut-off valve at the turbine inlet rather than a metering valve in order to increase the flow restriction size, and to be identical to the valve which has shown good service at this location for a period of approximately 4000 hours prior to that time.

d. Movement of the location of the centrifugal separator to a position immediately in from of the turbine and insulation of the liquid drain line from the separator to the interface in order to reduce the rate of condensation occurring in this line.

e. The present corrosion product separator at the boiler inlet was replaced by a single corrosion product separator in the liquid return of the boiler centrifugal separator in order to allow operation at temperatures of approximately 600°F to determine whether the corrosion product separator is functioning properly.

f. A check valve was added in the superheater centrifugal separator return line to prevent inadvertent back flow of fluid toward the turbine in the event of an emergency or normal shut down.

g. Incorporation of collector pots in various locations in the rig to determine whether corrosion products carried in the flow stream have a tendency to rise in vertical lines and allow trapping of these products. These collector pots were removal from the rig without effecting rig operation so they could be periodically inspected to determine whether the trapping of products in this manner was successful.

h. An inline filtering element was installed immediately in front of the turbine inlet housing to catch or restrict foreign particles.

When the unit was disassembled after its final test run the first stage nozzle area was measured and found to be the same as it was prior to the CSU I-3A-1 test which indicates the effectiveness of these rig modifications.

4.2.2 Start-Up and Speed Climb (0 to 20 Hours)

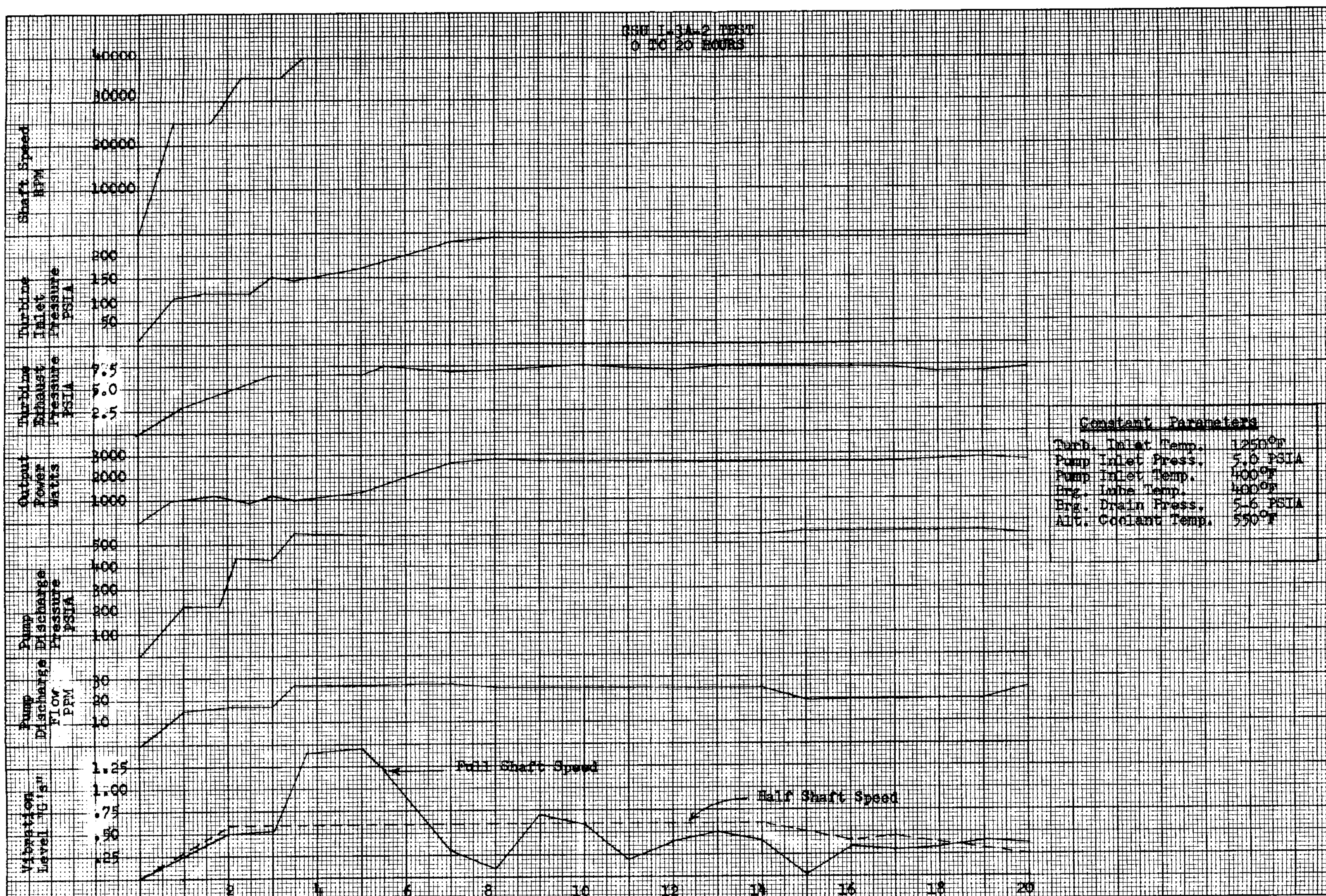
CSU I-1A-2 test was started at 1415 on April 16, 1963 Just prior to start-up the bearings were calibrated and the results agreed very closely with previous calibrations.

Turbine inlet pressure was gradually applied but the unit started before there was any pressure indication on the inlet pressure gauge. This was probably due to low system (turbine exhaust) pressure. The first speed plateau was to be 20000 rpm but the unit accelerated beyond this point and was controlled at 25000 in stead. Turbine inlet pressure was brought up to 100 psig. At this pressure the power was 1160 watts with a turbine exhaust temperature of 575°F. CSU pump discharge pressure was sufficiently high so the pump was valved to supply its own jet boost.

The speed was raised to 35000 rpm, again by-passing 30,000 because of first mode critical shaft speed. It was noted that all pressure readings less than atmospheric were at a very high vacuum. This was attributed to the system being very tight, therefore nitrogen was bled into the system to raise these pressures to their normal levels.

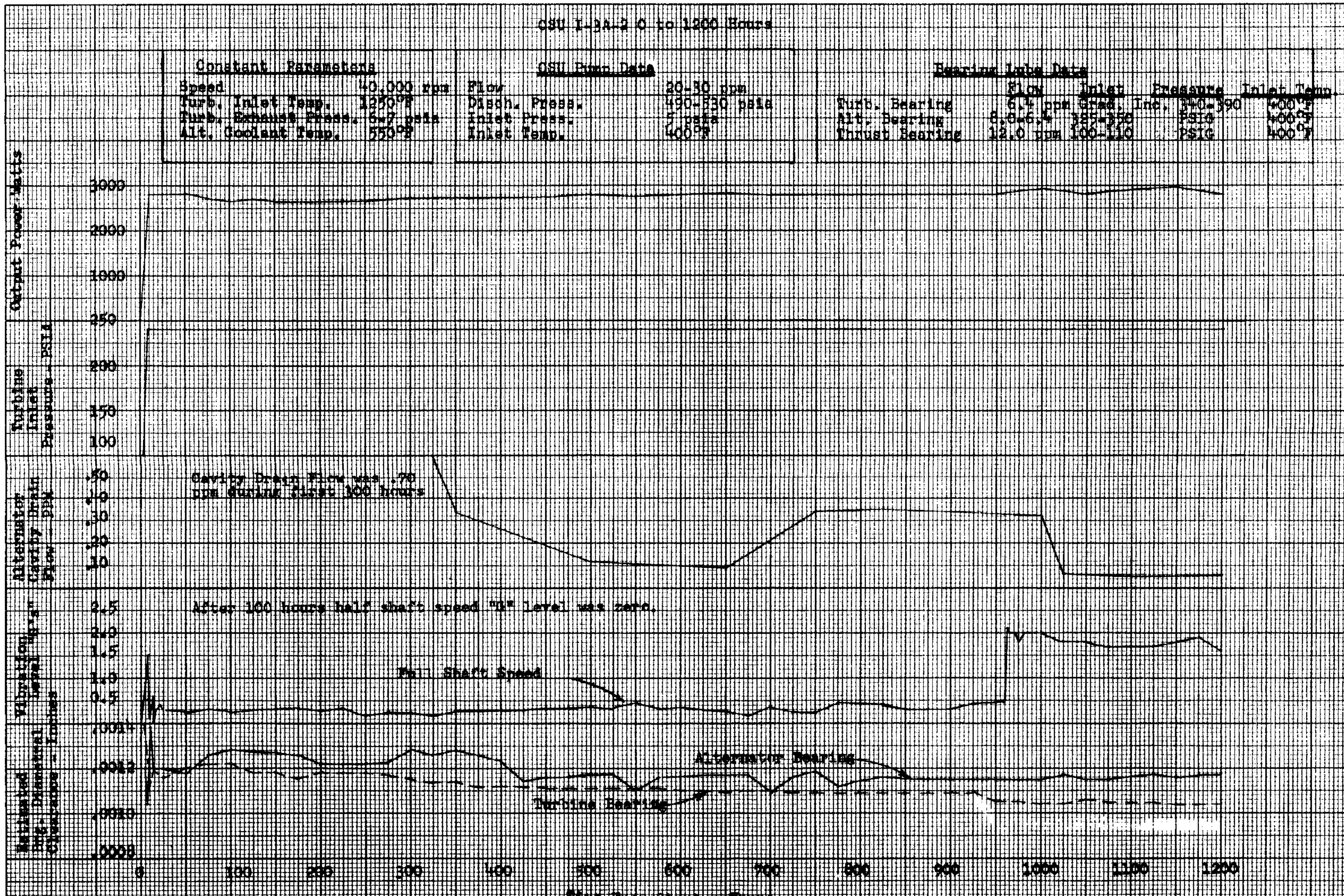
At an elapsed time of four hours the speed was brought up to 40000 rpm; output power was 960 watts, turbine inlet pressure was 140 psig, and the exhaust pressure was 7 psia. Over the next four hours the pressure was raised to 240 psia where the power output was 2900 watts.

Figure 9 is a plot of these first 20 hours of operation. As shown the turboalternator was operating at design conditions after 8 hours and by the end of this time period had essentially achieved steady state operation. There were the usual exceptions; the journal bearings and alternator cavity dran flow. Turbine bearing clearance fluctuated between .00115 and .00125 inches while the alternator bearing clearance fluctuated between .0012 and .0013 inches. This



Time From Start - Hours

FIGURE 9



CSU I-3A-2 0 to 1200 Hours
Constant Parameters
Speed 40,000 rpm
Turb. Inlet Temp. 1250°F
Turb. Exhaust Press. 6-7 psia
Alt. Coolant Temp. 550°F
CSU Pump Data
Flow 20-30 ppm
Disch. Press. 490-530 psia
Inlet Press. 5 psia
Inlet Temp. 400°F
Bearing Lube Data
Flow Inlet Pressure Inlet Temp.
Turb. Bearing 6.4 ppm Grad. Inc. 340-390 400°F
Alt. Bearing 8.0-6.4 325-350 PSIG 400°F
Thrust Bearing 12.0 ppm 100-110 PSIG 400°F
Output Power Watts
Turbine Inlet Pressure - PSIA
Alternator Cavity Drain Flow - PPM
Cavity Drain Flow was .70 ppm during first 100 hours
Vibration Level "g's"
After 100 hours half shaft speed "g" level was zero.
Full Shaft Speed
Estimated Brg. Diametral Clearance - Inches
Alternator Bearing
Turbine Bearing
3000
2000
1000
250
200
150
100
.50
.40
.30
.20
.10
2.5
2.0
1.5
1.0
0.5
.0014
.0012
.0010
.0008
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
Time From Start - Hours

FIGURE 10

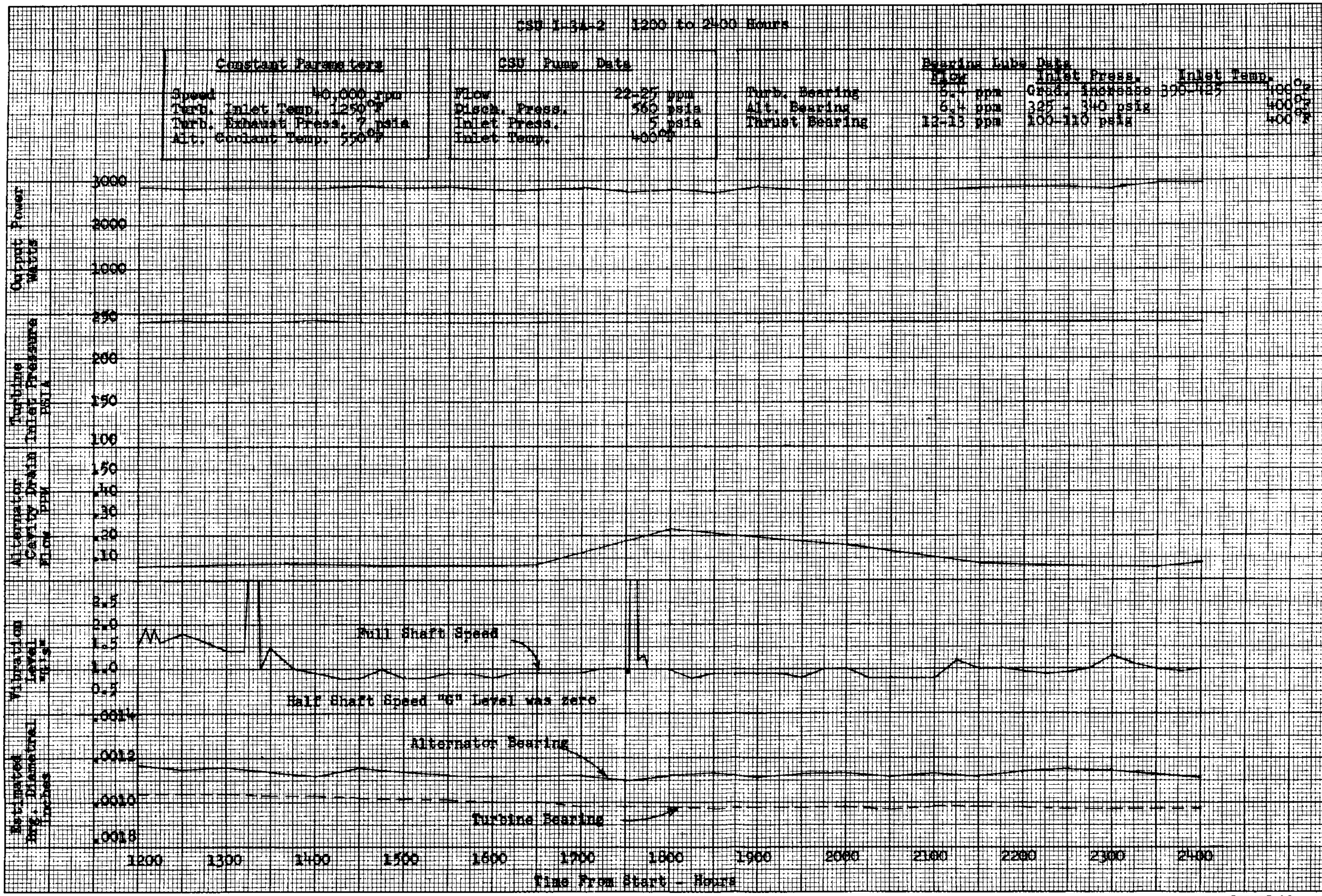
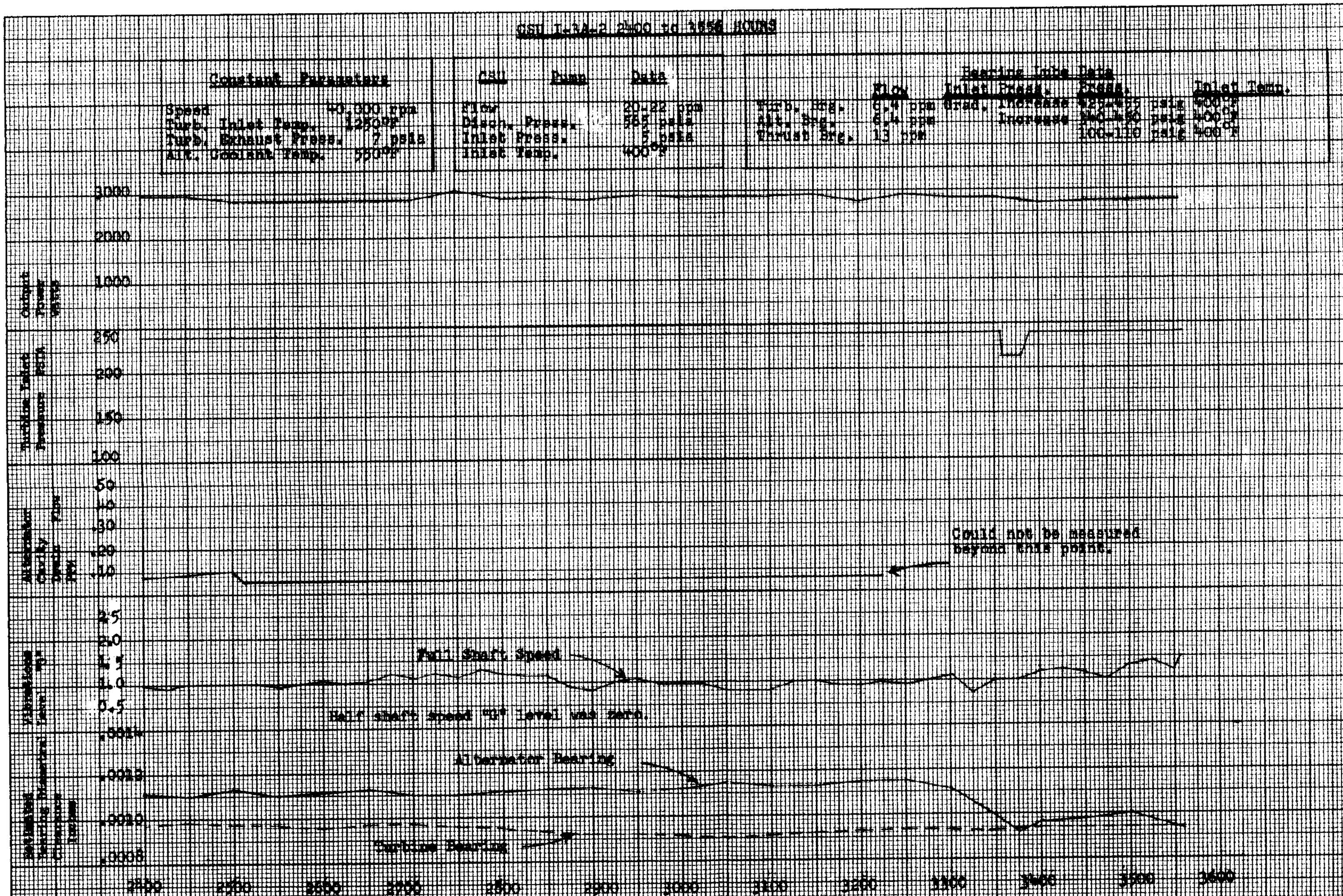

CSU 1-3A-2 1200 to 2400 Hours
Constant Parameters
Speed 40,000 rpm
Turb. Inlet Temp. 1250°F
Turb. Exhaust Press. 7 psia
Alt. Coolant Temp. 550°F
CSU Pump Data
Flow 22-25 ppm
Disch. Press. 560 psia
Inlet Press. 5 psia
Inlet Temp. 400°F
Bearing Lube Data
Flow
Inlet Press.
Inlet Temp.
Turb. Bearing
Alt. Bearing
Thrust Bearing
6.4 ppm
12-13 ppm
325 - 340 psig
100-110 psig
400°F
Output Power Watts
Turbine Inlet Pressure PSIA
Alternator Cavity Drain Flow PPM
Vibration Level "g's"
Estimated Brg. Diametral Inches
Full Shaft Speed
Half Shaft Speed "G" Level was zero
Alternator Bearing
Turbine Bearing
Time From Start - Hours

FIGURE 11



TIME FROM START - HOURS

FIGURE 12

time period to stabilize clearance was normal. With regard to the alternator cavity drain flow it will always be erratic until the newly designed dynamic seals are incorporated into the CSU. Two later units which have the new seals are assembled but have not been tested to determine the effectiveness of the new seal design.

4.2.3 Endurance Run (20 to 3556 hours)

Figures 10, 11, and 12 are time plots of several parameters for this 3556 hours endurance run. The parameters selected were those that were somewhat variable or had significant effect on turboalternator performance. Except for certain events near the end of the test the unit ran under steady state conditions with parameters constant.

Power output remained essentially constant at 2800 watts up until the last 200 hours when the alternator was accidentally parially demagnetized. This resulted in a power loss of 40 to 50 watts which is a relatively small amount considerating the operating conditions imposed on the unit. This occurrence will be discussed in more detail in a later paragraph. Turbine inlet pressure was maintained at 240 psia except for a short period following the alternator short. Turbine exhaust was controlled to 7 psia.

The following paragraphs discuss bearing and CSU performance, drain flow measurements and vibration level.

a. Bearings

Bearing performance during the CSU I-3A-2 test was nearly identical to that of the CSU I-3A-1 test except that it took longer for their clearances to stabilize. After about 350 hours the clearances were holding steady at .0011 in. for the turbine bearing and .00115 in. for the alternator bearing. Thrust bearing flow was 12 ppm at 105 psig. Half speed whirl was in evidence during the first 100 hours with shaft speed "g" level running at .35 "g's" and half speed at .2 "g's". After 100 hours the "g" level was barely measurable and after 350 hours it was nonexistent.

When the unit first reached 40,000 rpm the bearing flows were as follows: Thrust bearing, 12.5 ppm at 110 psig; alternator bearing, 6.5 to 7.0 ppm at 315 psig average; turbine bearing, 6.5 to 7.0 ppm at 295 psig average. The thrust bearing flow was maintained at this level with only minor adjustments during the test. Shortly after reaching 40,000 rpm both journal bearings were set at 6.4 ppm which was to be their nominal flow rate for the CSU I-3A-2 test.

During the first 70 hours, however, several attempts were made to suppress half speed whirl by adjusting journal bearing flows. The turbine and then the alternator bearing flows were increased to 7 ppm with very little effect on whirl. Alternator bearing flow was lowered to 6 ppm and then returned to 6.4 at which time the turbine bearing flow was reduced to 6 ppm. Again none of these changes reduced whirl. Flows to the turbine and then the alternator bearings were raised to 7.5 ppm which decreased whirl but only temporarily. At the 70 hour mark the alternator bearing flow was raised to 8 ppm and turbine bearing flow to 7 ppm decreasing whirl to practically zero. These flows were maintained at these values for an additional 260 hours in the case of the turbine bearing and 300 hours in the case of the alternator bearing at which time they were returned to their normal 6.4 ppm. Whirl returned to its original magnitude at 70 hours but dropped to a very low value at 100 hours and disappeared at 350 hours. It was at this same time that the clearances stabilized.

Thrust bearing performance remained constant throughout the test except for a period between 1200 and 1800 hours when pressure fluctuations were noted (±10 psi). The flow was gradually raised to 13 ppm where it remained for the balance of the test. This was very likely related to instability of the alternator bearing which also was occurring at that time. This instability was the result of fluctuating alternator cavity drain flow which was difficult to control during this time period.

In general the alternator bearing performance was satisfactory with its diametral clearance remaining in the .0011 to .00115 inch range up until the final 250 hours. At that time it dropped to .0009 inch, gradually returned to .0010 and stayed to the end of the test. Except for brief periods of steady operation the alternator bearing supply pressure was somewhat erratic throughout the test with fluctuations of ±15 psi. These pressure fluctuations appeared to be storngly influenced by alternator cavity drain flow. This problem should be eliminated by the new dynamic seal between alternator bearing journal and the alternator cavity.

Prior to the decrease in alternator bearing clearance near the end of the test, a leak had developed in a mercury to water heat exchanger allowing water to enter the test rig. It is unknown whether water actually traveled to the bearing cavity but the leak was repaired without stopping the test and subsequently the clearance slowly returned towards its original value.

Although the clearance never competely returned to its original value and because of later seizure of the bearing it was impossible to determine if the clearance had actually closed down.

The turbine bearing clearance had a general tendency to decrease throughout the test. The minimum clearance during the test was .0009 inches. Its final value at shut down was .00094 which was identical to the final alternator bearing clearance. Several attempts were made to adjust operating conditions to stop the downware trend in clearance but none was actually successful.

These adjustments, in the order in which they were attempted, areas follows. (Note: In each case conditions were returned to normal when it was found that the change had no effect).

a. Turbine bearing lube temperature was lowered to 375°F and raised to 425°F.
b. Turbine exhaust temperature was lowered to cool the shaft and thus reduce the journal diameter.
c. Bearing supply pressure was raised to 450 psig.
d. Turbine inlet pressure lowered to 225 psia and raised to 255 psia.

Although none of the above parametric changes checked the downward trend in indicated clearance, the bearings were always within the design specifications for flow, pressure and clearance. At disassembly the turbine bearing had the identical clearance as originally measured. A thin film found deposited on the bearing was apparently responsible for the indicated change in clearance. This film will be discussed further in the disassembly section of this report.

B. Mercury Pump

Initially the pump performance was the same as at the start of the CSU I-3A-1 test (26 ppm at 560 psia). However, during the first 170 hours output pressure dropped to 520 psia at 26 ppm. During the next 300 hours the flow was gradually increased to 30 ppm where the pressure was 495 psia. Flow and pressure gradually increased to 25 ppm and 560 psia at 1600 hours which is the same performance as at the start of the test. Shortly therafter the flow was purposely reduced to 22 ppm and ultimately to 20 ppm.

Pump performance at the end of the CSU I-3A-2 test was identical to the initial performance of the CSU I-3A-1 test. In fact it was slightly improved as shown in Figure 13. The curves compare calibrations before and after a span of 3815 hours operating time. This performance is well above design and is especially significant since severe erosion of the pump impeller was noted during disassembly. The condition of the impeller is discussed in section 7.1.

C. Drain Flow Measurements

Measurements of the four drain flows (combined alternator and thrust bearing, turbine bearing, turbine vapor by-pass, and alternator cavity) were made during the CSU I-3A-2 test in the same manner as during the CSU I-3A-1 test. Figure 6 shows these drain flows as a function of time for the CSU I-3A testing. As shown by these curves the flows remained relatively constant with the exception of the alternator cavity drain. As was the case for the CSU I-3A-1 test the bearing flows were used as a continuing check on bearing flow venturi accuracy and bearing performance.

The information obtained during the CSU I-3A-1 test concerning the alternator cavity drain flow, its control, and its effect on CSU performance was verified in the CSU I-3A-2 test. However, control of the flow was more difficult although the same methods were used in both tests. In order to limit the cavity flow within certain values it was necessary to keep the alternator bearing drain pressure within the range of 20 to 22 inches of mercury vacuum. This was accomplished by varying the CSU pump flow and opening or closing the alternator bearing degas valve. Lowering the pump flow lowered the pressure and opening the degas valve, which referenced the bearing drain pressure to condenser pressure, raised the pressure.

As shown in Figure 6 the alternator cavity drain flow remained at a high level during the first 300 hours despite continuous efforts aimed at reducing the flow by varying pump flow and turbine exhaust pressure. The power output of the CSU was correspondingly low.

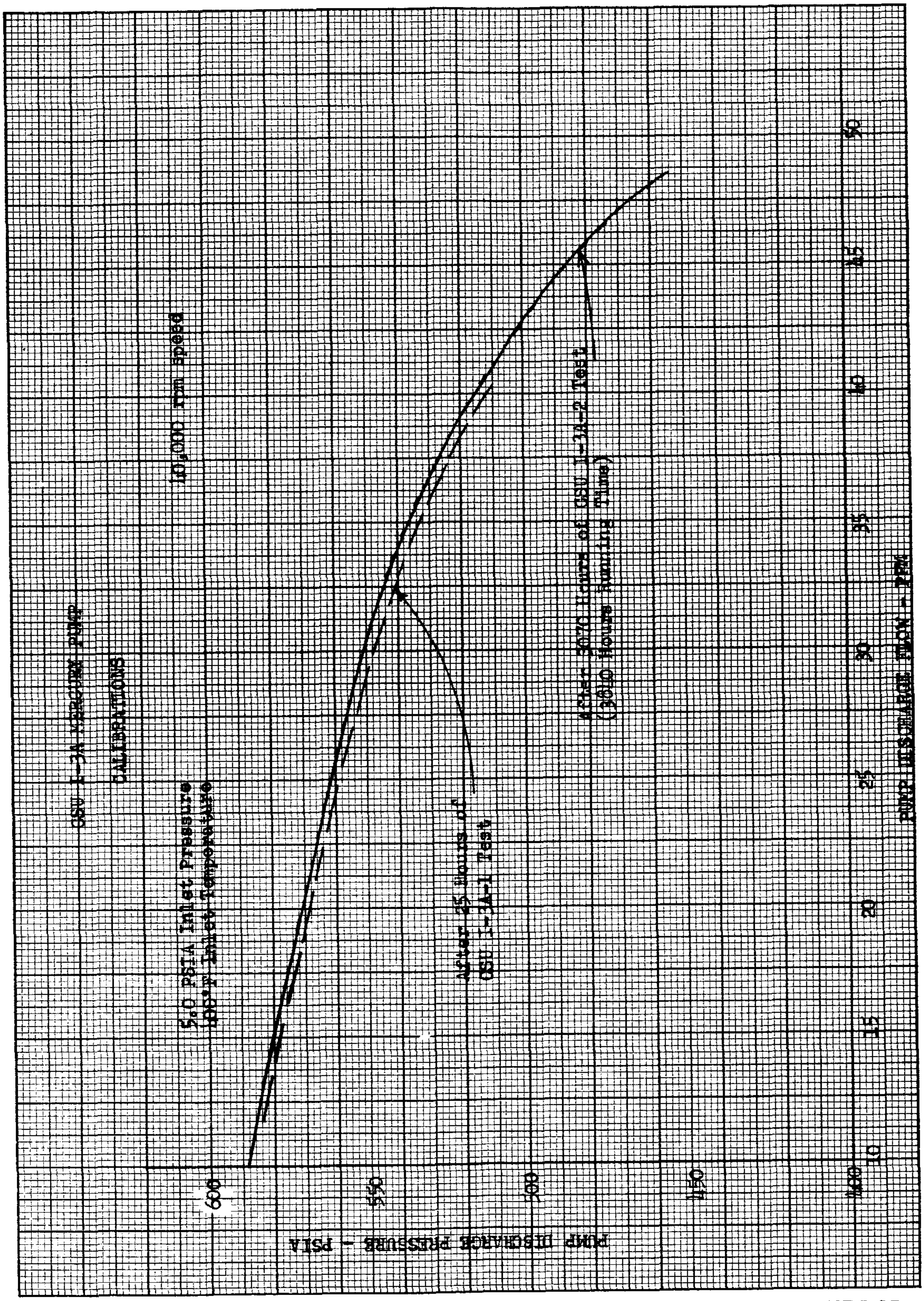

OSU I-3A MERCURY PUMP
CALIBRATIONS
10,000 rpm speed
5.0 PSIA Inlet Pressure
100°F Inlet Temperature
After 25 Hours of
OSU I-3A-1 Test
After 3070 Hours of OSU I-3A-2 Test
(3010 Hours Running Time)
PUMP DISCHARGE FLOW - GPM
PUMP DISCHARGE PRESSURE - PSIA
10
15
20
25
30
35
40
45
50
600
550
500
450
400

FIGURE 13

Shortly after 300 hours the pump flow was lowered to 20 ppm which resulted in a 50% reduction in cavity flow and a gain of 65 watts in output power. Gradually the cavity drain flow was lowered to .10 ppm. Subsequently it climbed to .35 ppm and remained there until an elapsed time of 1000 hours when it was reduced to .08 ppm. This value corresponded to the flow rate experienced during the CSU I-3A-1 test. At this time the output power was 2800-2850 watts where it remained for the balance of the test.

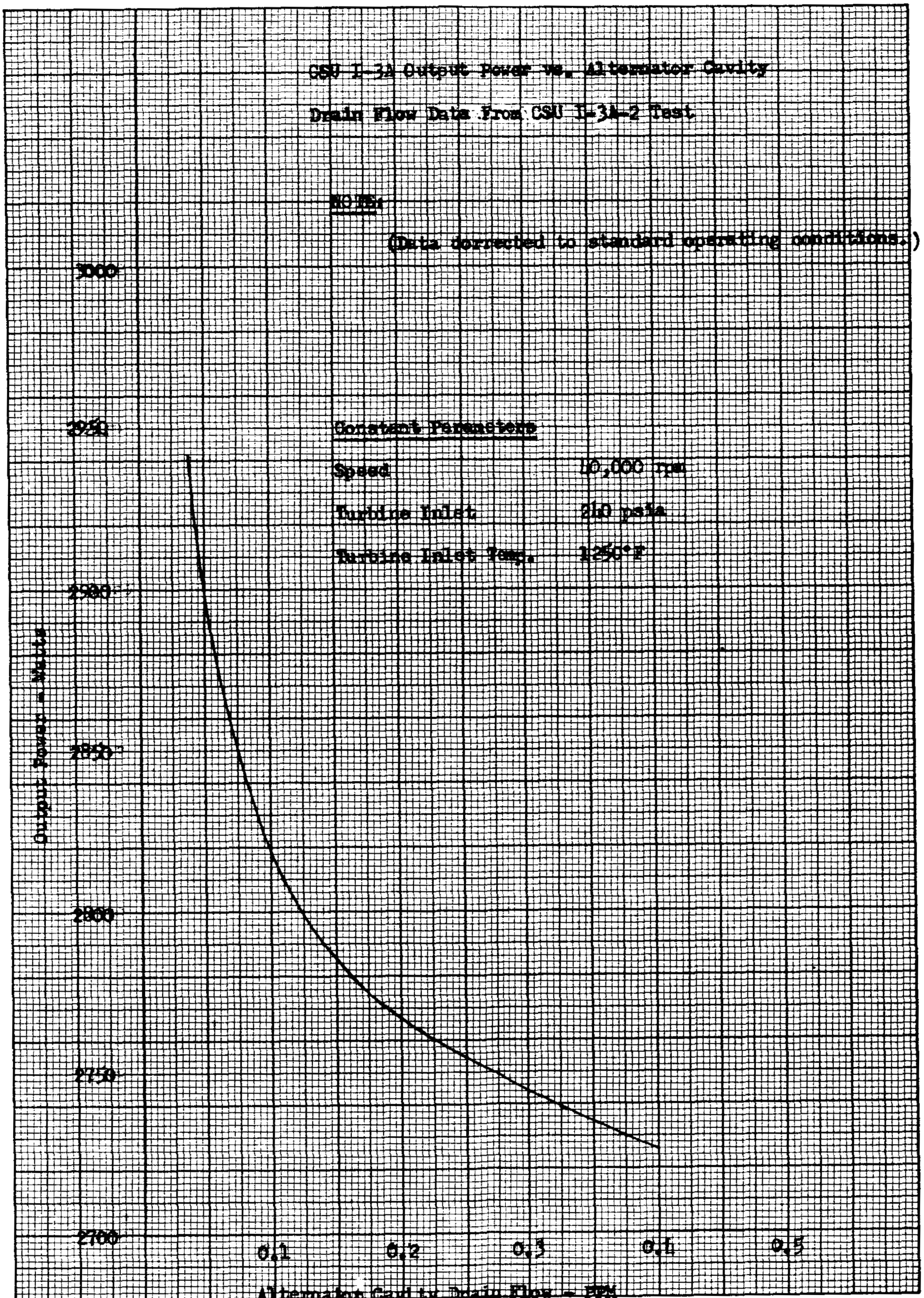
After 1000 hours pump flow was maintained in the 20 to 25 ppm range and the alternator bearing drain degas valve partially opened. This maintained the bearing drain pressure in a range which kept the alternator cavity drain flow at a low level. There were occasions when the cavity drain flow could not be controlled resulting in CSU output fluctuations. Figures 14 and 15 are plots showing the relationship of cavity drain flow vs. temperature and the relationship of output power to cavity drain flow. These curves are nearly identical to those obtained during the CSU I-3A-1 test.

D. Vibration

The nominal vibration level during this test run was one "g" with several occasions when it deviated from this value. In the previous discussion of the bearings it was indicated that half speed "g" level was present for only the first 300 hours. When measurable, the vibration at half shaft speed was .1 to .2 "g" except during the initial 20 hours when it was .6 "g's".

There were four occasions when shaft speed vibration level exceeded 1.0 "g" and are as follows:

1. The "g" level for five hours after start-up was between 1.25 to 1.5 which is a normal occurence while the unit is achieving stable operation.
2. At 960 hours the "g" level jumped to 2.0 and then gradually de-



CSU I-3A Output Power vs. Alternator Cavity
Drain Flow Data From CSU I-3A-2 Test
NOTE:
(Data corrected to standard operating conditions.)
Constant Parameters
Speed 10,000 rpm
Turbine Inlet 240 psia
Turbine Inlet Temp. 1250°F
3000
2950
2900
2850
2800
2750
2700
Output Power - Watts
0.1
0.2
0.3
0.4
0.5
Alternator Cavity Drain Flow - PPM

FIGURE 14

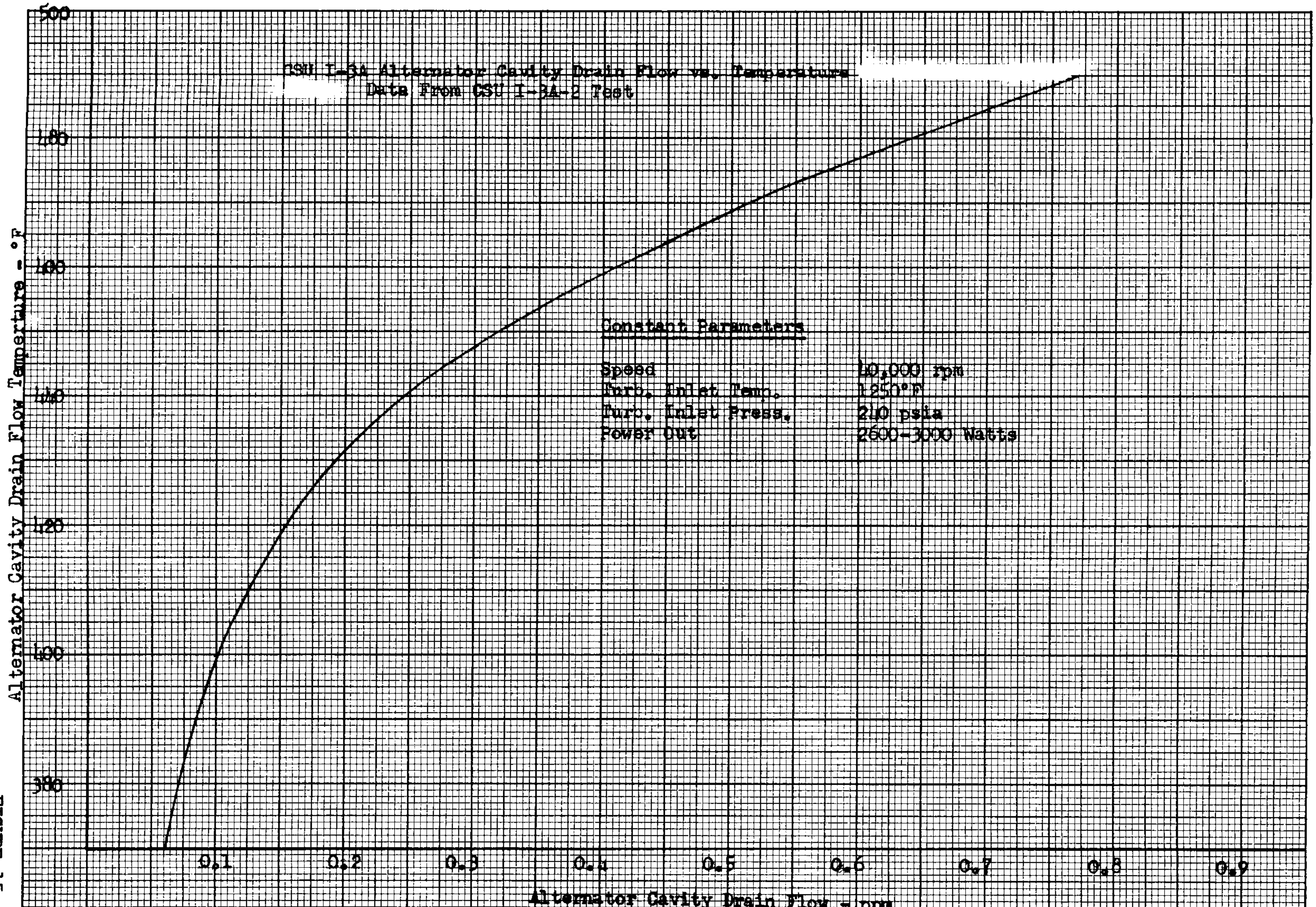

CSU I-3A Alternator Cavity Drain Flow vs. Temperature
Data From CSU I-3A-2 Test
Constant Parameters
Speed 10,000 rpm
Turb. Inlet Temp. 1250°F
Turb. Inlet Press. 210 psia
Power Out 2600-3000 Watts
500
480
460
440
420
400
380
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
Alternator Cavity Drain Flow - ppm
Alternator Cavity Drain Flow Temperature - °F

FIGURE 15

clined to 1.5 "g's" at 1320 hours.

3. At 1320 hours the vibrations increased to 4.0 "g's" which was verified with a portable accelerometer. The level gradually dropped to 1.0 during the next 20 hours.
4. At 1750 hours the vibration increased to 2.6 "g's" which was also verified with a portable accellerometer. Eight hours later it was back at 1.0 "g's" where it remained for the balance of the test run.

A careful review of the test data and operating conditions at the time of each of the three sudden jumps in "g" level did not reveal any explanation for these occurrences. The only clue came at disassembly of the CSU which disclosed three cracks in the alternator rotor which could have been responsible. These cracks were probably slight when they first occurred which propogated into fractures during the overspeed at final CSU failure.

E. General Comments

The preceeding paragraphs discussed most of the test results but there were additional occurences of interest which affected the operation of the unit. Most happenings concerned interruption of plant services and minor test rig malfunctions. These events will be discussed in the order in which they occurred.

At about 1300 hours elapsed time 440 volt power failed instantaneously. This is the power supply for the test rig mercury pump, boiler, and superheater. When the power was restored the auxiliary rig pump took over which requires some adjustments, the boiler and superheater controls had to be reset, and the system vacuum pump had to be reset. While this was taking place turbine inlet pressure dropped to 125 psig and speed to 29900 rpm. Conditions were gradually returned to normal and the unit was performing as well as, or better, than before the power failure.

Approximately 40 hours later the overspeed switch shorted which closed the turbine inlet valve reducing turbine inlet pressure to zero. Speed had dropped to 10100 rpm when a manual override switch was closed which opened the turbine inlet valve. The speed immediately climbed to 40,000 rpm and operation returned to normal.

At an elapsed time of 1360 hours it was decided to use the CSU pump to supply the bearing lube flows. This was deemed to be desirable at the time because bearing lube flow could be maintained if the test rig pump failed and the auxiliary pump wouldn't start. Sufficient mercury was available in the CSU pump inlet reservoir to supply the bearings for about 2 minutes which would allow enough time to take corrective action or shut the test down. The change over from CSU pump to rig pump bearing supply was made without incident and the system valving was such that the rig pump would supply flow if the CSU pump failed.

After about 4 hours of operation in this manner it was noted that the pump flow was less than the total bearing flows which indicated a leaking check valve or erroneous reading of the pump flow venturi. Therefore, the system was returned to rig pump supply for the bearings.

Two days later an accumulator was added to the bearing supply line so that in case both the main and auxiliary rig pump became inoperative (such as due to a power failure) there would be a source of lube flow for the bearings until the test was shut down. This system was capable of providing a minimum of 250 psig at full bearing lube flow for at least five minutes.

At 1410 hours a second 440 V power failure was experienced with the same results except that speed decreased to 35,000 rpm and turbine inlet pressure to 145 psig. Again the CSU was returned to normal operating conditions with no change in performance.

On July 9, 1963 it was decided to explore the possibility of operating the test unattended. The unit had been operating for over 2000 hours during the test run and an accumulated total of nearly 2800 hours with no indication that it could not continue to run indefinitely. The intent was to eliminate around the clock coverage by test personnel if proven feasible.

It was recognized that certain parameters would have to be controlled during this "hands-off" operation but the rest would be allowed to drift unless certain limits were exceeded. The parameters to be controlled were condenser pressure, boiler level, and alternator bearing drain line pressure. Boiler level and condenser pressure could be controlled automatically but the alternator bearing drain line pressure, which directly influences alternator cavity drain flow, could not be controlled and would have to be allowed to seek its own level. The "hands-off" run was made to find out whether the rest of the test parameters would remain within acceptable limits.

During the first 30 hours, operation was steady and smooth with adjustments required for only the three specified parameters. Turbine inlet pressure and bearing flows remained within specification while bearing lube temperatures averaged 10°F and pump inlet 15°F above specifications, neither of which was unacceptable. However, during the balance of the "hands-off" run the following additional adjustments were required:

1. Boiler power.
2. Thrust bearing pressure.
3. Alternator coolant flow.
4. Pump inlet temperature.
5. Turbine bearing heater.
6. Superheater power.

After 147 hours the "hands-off" operation was discontinued as adequate information was obtained to establish requirements for automating the test if a decision had been made to do so. When the test conditions were returned to their usual values CSU performance was found to be unchanged.

At an elapsed time of 2280 hours, water supply to the test rig was interrupted for 35 minutes. Since this water is used as a coolant for the test rig pump and several subcoolers this could have resulted in a test shut down. Fortunately no harm resulted to the unit although turbine exhaust temperature did climb to 645°F and the main rig pump failed with the auxiliary pump taking over.

About 120 hours later the thrust bearing lube heater cut out allowing the lube temperature to drop to 275°F which caused the inlet pressure to climb to 150 psig from its usual 105. When the heater was re-energized the parameters returned to normal.

The next day a circuit breaker cut out which fed the alternator coolant, superheater discharge, and turbine housing heaters. Turbine inlet temperature dropped from 1260 to 1220°F, turbine housing heater from 1230 to 1090°F, and alternator coolant from 550 to 430°F. The breaker was out for 5 to 6 minures but it took only 15 minutes for these temperatures to return to normal.

There was a period between 2800 and 3350 hours when it was difficult to measure the alternator cavity drain flow because the fluctuations, although not severe, came at close intervals which didn't allow sufficient time for an accurate measurement. At 3250 hours, however, the flow did remain steady long enough to obtain a reliable value. For the given set of operating conditions and drain flow at that time, the CSU power output compared very favorably with comparable data taken several months earlier. This fact leads to several observations concerning the test operation. Corrosion product deposition in the turbine inlet housing had apparently been prevented. Two things were mainly responsible for this; the rig modifications made between the CSU I-3A-1 and the CSU I-3A-2 test and the fact that the turbine inlet housing heater was operating throughout the test. It is also indicative of the turboalternator's ability to withstand the suddenly imposed off-design conditions caused by the interruped plant services and test rig malfunctions.

Prior to the municipal power failure there were two more significant events which appreciably affected CSU performance and operation. These events included a malfunction of the bearing drain line subcooler and a partial demagnetization of the alternator rotor.

1. Malfunction of Bearing Drain Lines Subcooler

A leak developed in the mercury to water heat exchanger used in subcooling the bearing drains and alternator and turbine vapor by-pass cavities. This allowed water to enter the mercury test rig where it was vaporized and carried to the condenser. The influx of steam to the rig caused the condenser pressure to fluctuate considerably.

The water vapor was removed by the use of vacuum pumps in the test rig. Occasionally, while maintaining a vacuum on the system, traps located between the condenser and vacuum pump had to be opened and drained of accumulated water. During this phase of operation turboalternator back pressure was raised to atmospheric conditions resulting in turbine exhaust temperatures of 675°F.

Shortly thereafter the alternator bearing pressure increased approximately 100 psi while constant flow was maintained. It is unknown whether water actually traveled to the bearing cavity. However, it seems more than coincidental that the pressure would change at the time the leak developed since it had previously operated 6400 hours with only minimal change. Throughout the following week the bearing pressure slowly returned toward the original value further indicating some effect due to the entrance of water into the test rig.

A short time after the subcooler leak was discovered it was decided to continue the test and make repairs without stopping the unit. The method chosen to eliminate the water leakage was to use mercury instead of water as the coolant. Thus any leakage within the subcooler would not affect the test rig or the turboalternator unit.

It was found after making the necessary connections that the mercury to mercury heat exchanger was not effective enough at the flow and temperature conditions available. To compensate for this an auxiliary cooling loop was wrapped around the subcooler.

2. Partial Demagnetization of the Alternator Rotor

During the morning of September 3, 1963 the auxiliary coolant was provided by wrapping copper tubing around the mercury line leading to the drain manifold subcooler. However, during the installation, one of the copper lines contacted the alternator short circuit capacitor hot lead. The arcing which resulted partially demagnetized the alternator rotor from a voltage level of 110 volts to approximately 70 volts. Since this effectively decreased the load as seen by the alternator the unit speed increased to 43,000 rpm. It was controlled by adjustment of the manual load and reduced to 40,000 rpm. Although no apparent damage occurred to the unit it was recognized that to maintain a fixed load and constant speed an increased amperage would be required in each phase. Until a decision could be made and co-ordinated with NASA, it was decided to lower the turbine inlet pressure to a value which maintained the alternator current at approximately 17 amps per phase. This was accomplished and the unit output power at this point was approximately 2100 watts. The lower voltage level which existed necessitated several changes in test rig electrical equipment. These changes were as follows:

a. Step-up transformers were installed on the input side of the speed control to raise the voltage into the speed control from 70 to 110. This was accomplished to allow the control to operate at its design voltage.
b. Current transformers were connected to the wattmeters and ammeters in the rig consoles to double their range. Thus, they would be capable of reading the higher currents to be encountered at full turboalternator power.
c. The increased current that would result from return to full power was expected to raise the temperature at the electrical connector. To monitor this temperature while power was returned to normal, a thermocouple was installed on the connector barrel. The temperature indication from this thermocouple was continuously recorded on a logger.

After observing all the data and obtaining NASA's approval it was decided to bring the unit to full power under the above conditions and allow the alternator electrical connector temperature to assume a value up to a maximum of approximately 350°F. At the resumption of full power operation, this temperature read 320°F with variations of ± 10°F. It was felt at this time that the conditions was satisfactory and the unit was allowed to continue. The unit ran approximately 1 week at these conditions without incident.

4.2.4 Shut Down

At 18:09 September 11, 1963, a plant wide power failure occurred which was traced to a substation supplying the plant from the municipal power station. The power was off for approximately 4 minutes, but since the duration was uncertain at the time, action had to be taken immediately to attempt to preserve the unit from damage. After approximately 30 seconds of operation with no power to the facility a normal emergency shut down procedure was followed. The basic steps involved in this procedure are manual closing of the turbine inlet valve, closing of other assorted valves throughout the system and the deactivation of all power switches. By the time the actual power was restored the turbine inlet valve, boiler inlet discharge valve, superheater drain v alve, and chempump by-pass valve had been closed. The emergency bearing supply accumulators had been supplying flow to the bearings with their inventory being discharged into the test rig.

Subsequent bearing calibrations were conducted on the unit which varified that the unit had been successfully stopped without sustaining damage.

4.3 CSU I-3A-3 Test

Since the CSU appeared to have sustained no damage due to the shut down of the I-3A-2 test it was decided to restart the unit as soon as possible. However, there were several test rig corrections made to alleviate some problem areas which had occurred in the rig previously. These included the following.

a. The stand-by chempump was replaced.

b. The test rig mercury inventory was adjusted to correct for the recharging of the bearing accumulators and to replace the mercury which was lost from the system during rig modifications.

c. Insulation was removed from the turboalternator unit for observation of possible leaks in the turbine area. Lower layers of the insulation appeared to be free of mercury and the unit was reinsulated to its original thickness.

d. A water subcooler was added to the bearing drain manifold coolant line.

4.3.1 Start-Up and Speed Climb

After the preparatory work had been completed the CSU was preheated and readied for restart. At 06:14 on 9-12-64, approximately 12 hours after the initial shut down, the unit was restarted and brought up to full speed and power over a period of approximately 3.5 hours. It was at 25,000 rpm for approximately one hour and at 35,000 rpm for about an hour. At full speed and turbine inlet pressure the power output was 2820 watts. All three bearings returned to the same conditions at which they were operating just prior to the last shut down.

4.3.2 Shut Down

The turboalternator unit had been at full turbine inlet pressure and output power for approximately 15 minutes with satisfactory performance. Suddenly, without warning, a short occurred in the alternator test rig connector which demagnetized the permanent magnet rotor and allowed the shaft speed to increase from 40,000 to approximately 49,600 rpm. Full manual load was applied in an attempt to control speed. Although the reduced voltage resulting from the demagnetization of the rotor decreased the capacity of the load bank it apparently was still possible to load the turboalternator sufficiently to maintain control.

After about twenty seconds of operation at 49,600 rpm (at which time the manual load was applied) the speed gradually reduced to about 40,000 rpm. This was followed by a sharp decline in speed which included rapid accellerations and decellerations. Vapor flow to the unit was stopped by actuating the turbine inlet solenoid valve. Shaft speed dropped to zero and bearing flows were readjusted to their specified values. It was decided to follow the normal shut down procedure but due to the confusion during shut down the closing of the turbine inlet valve was misinterpreted. This valve utilizes air pressure to close it and inadvertently the air supply had been removed. Therefore this valve, which was previously closed, reopened allowing full turbine inlet pressure to be exerted on the turbo-alternator unit. Shaft speed increased rapidly and the unit ran erratically. Later examinations of the dynamic recordings indicate fluctuations in shaft speed between 20,000 and 56,000 rpm with the upper value being limited by pen travel. After several minutes of operation in this mode the unit experienced complete rotational stoppage. Approximately three minutes after shaft stoppage the turbine inlet flow was shut-off by a valve located at the boiler discharg. A reproduction of the transient data from the recording oscillograph is shown in Figure 16 .

The following summary outlines timewise the occurances from initial overspeed to shaft seizure. A total of 8 minutes elapsed from the first indication of CSU performance variation until the final boiler flow was shut-off. This time period is conveniently divided into the five parts.

a. Two minutes and twenty seconds.

The short in the electrical connector occurred at the start of this time period. This was evidenced by the sudden reduction in alternator output and climb in speed. Also the temperature of the connector shell rapidly rose to 460°F from its normal level of 350°F.

As a result of the demagnetization the output voltage fell from 75 to 40 volts. (A completely demagnetized alternator rotor has about 40 volts remaining due to its residual magnetism).

Parasitic current dropped to zero because the speed control was no longer functioning since it senses phase one which was shorted. Phase 2 total load amps fell from 21 to 15 amps because the bulk of the output was probably flowing through the short in phase one.

Turbine inlet temperature and pressure (and consequently flow) remained constant throughout this period; thus the shaft input power did not change. Since the alternator was producing a lower voltage and load bank was absorbing less power. With the speed control not functioning it was necessary to adjust the load bank to 100% to try to return the speed to 40,000 rpm. This, in combination with the short in phase one, gradually brought the speed down to normal.

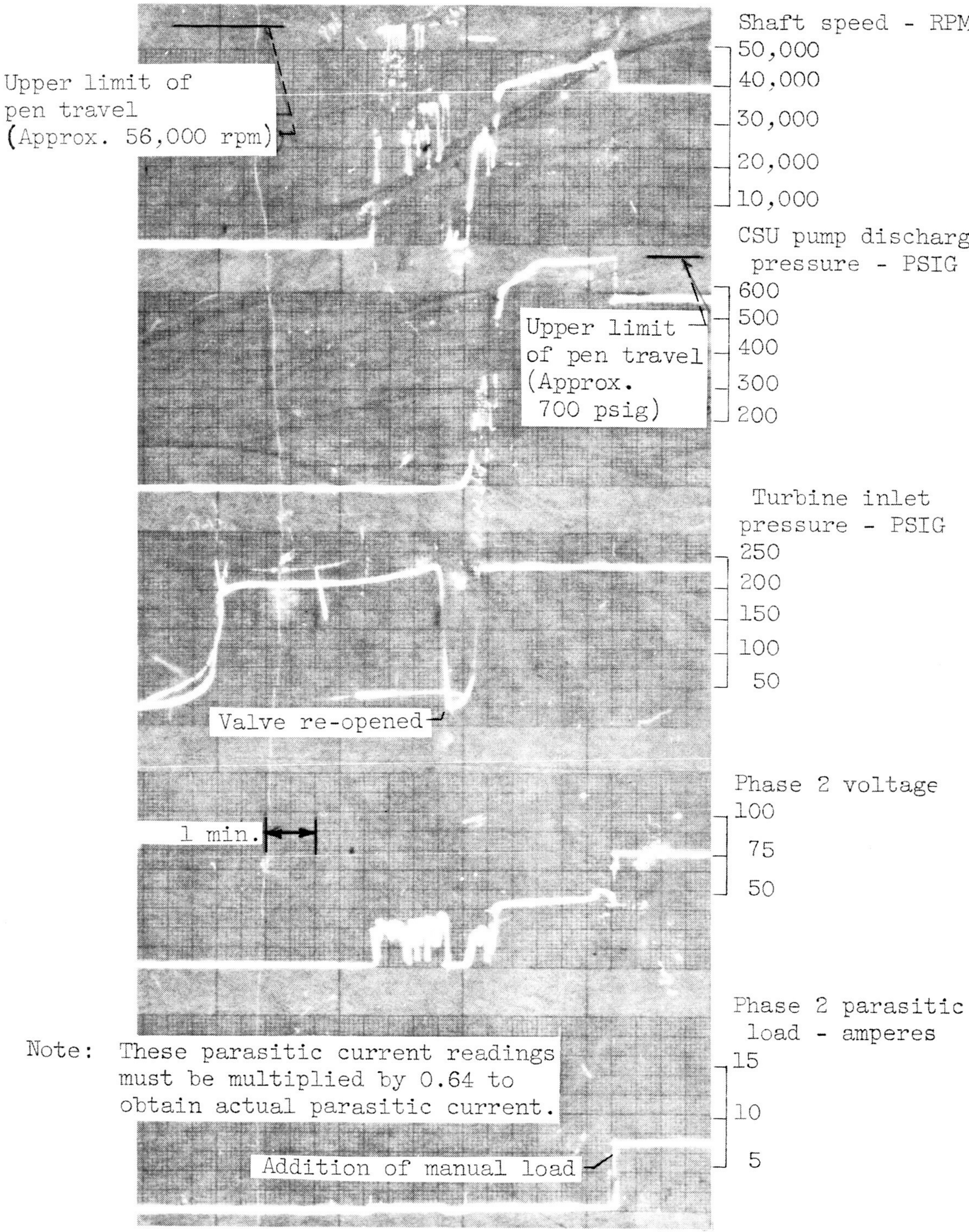
Bearing flow and pressures fluctuated due to the speed changes but the variations were what would be expected. Since both journal bearings were operating at the maximum available supply pressure no change in pressure was noted. However a decrease in flow did accompany the higher speed. The turbine bearing flow dropped from 6.1 to 4.4 ppm and returned to 6.1 at a constant 450 psig inlet pressure. The alternator bearing reacted much the same way with a decrease from 5 to 3 ppm and a return to 4, all at 450 psig inlet. Thrust bearing flow didn't vary from its 12 ppm level but the pressure fell from 110 to 90 and returned to 110 psig.

b. Thirty Seconds

During this time period shaft speed was very erratic with fluctuations between 40,000 and 25,000 rpm. At the end of the period speed was zero since vapor flow to the turbine was turned off. The likely reason for the speed fluctuations was interference between the alternator rotor and stator. The high temperatures generated within the alternator, because of the short, probably caused melting of the glass bore seal. All housing temperatures in the vicinity of the alternator rose during this time period as did the alternator coolant outlet temperature and the electrical connector shell temperature.

Phase 2 voltage fluctuated between 0 and 40 volts, phase 2 total current between 0 and 15 amps. These variations corresponded to the speed change characteristics. With the speed control not operating (because the short was in phase one) the parasitic current was constant at zero.

Bearing operation seemed to be normal during this period with



RECORDING OSCILLOGRAPH RECORD OF FINAL MINUTES OF SUNFLOWER
CSU I-3A-3 TURBOALTERNATOR TEST

FIGURE 16

the journal bearing flows rising and pressures falling as speed decreased. In the case of the thrust bearing, where the pressure is more dependent on shaft axial position than on speed, the pressure increased as flow decreased. When the turboalternator speed was in the 25,000 rpm range the turbine bearing flow was up to 12 ppm from a normal 6.1 while the pressure fell from 450 to 420 psig. The alternator bearing flow also rose to 12 ppm at this speed level and pressure fell to 380 psig. Thrust bearing flow remained constant at 12 ppm while pressure tended to increase with decreasing speed.

The turbine inlet solenoid valve was closed a few seconds before the end of this time period which resulted in a rapid decay in turbine inlet pressure and speed. At the end of the time period the unit was at zero speed.

c. Thirty Seconds

The turboalternator was at zero speed during this time period because of the closed turbine inlet valve. Temperatures in the turbine area started to decrease appreciably with the absence of vapor flow while the temperatures in the vicinity of the alternator tended to continue rising gradually, the alternator connector shell temperature increasing from 500 to 510°F.

CSU pump discharge pressure and all electrical output parameters were zero because of the zero speed. Again there was no change in thrust bearing flow but the inlet pressure rose to 140 psig at zero rpm as compared to 115 psig at normal speeds. Turbine bearing flow rose from 12 ppm to 17.4 ppm at zero speed while the inlet pressure fell from 420 to 315 psig. Alternator bearing flow increased 12 ppm to a value of 18.5 ppm at zero speed with supply pressure decreasing to 190 psig from 380 psig. Because of the short time interval involved, conditions didn't reach equilibrium and thus no estimation of bearing clearances could be made at the off-design conditions.

At the end of this time period the turbine inlet valve was inadvertently reopened resulting in full turbine inlet pressure again being supplied to the turboalternator causing it to climb rapidly in speed.

d. One Minute and 30 Seconds

During this time period the unit accelerated from zero speed to at least 56,000 rpm which is the limit of the pen travel on the recording oscillograph. Throughout this period the speed was very erratic, fluctuating between 20,000 and 56,000 rpm. Turbine inlet pressure started out at 225 psig and gradually tapered off to 210 at a temperature of 1250 to 1260°F.

Phase 2 total current fluctuated between 0 and 10 amps and phase 2 voltage between 0 and 40 volts. Parasitic current remained at zero since the speed control was not operating.

All three bearings reacted sharply to the speed fluctuations. Even the thrust bearing, which had been relatively insensitive to previous excursions, climbed in inlet pressure from 115 to 195 psig. There were some very slight fluctuations in flow but it still remained essentially at 12 ppm. Alternator bearing flow was very erratic, varying between 18.5 and 3.5 ppm. Its supply pressure climbed from 190 to 380 and then fell to 60 psig. The wide variations in flow and pressure were due to the speed fluctuations and to adjustments by the test personnel attempting to control bearing flow. Turbine bearing flow and pressure was also very erratic with flow fluctuating 17.4 and 11.2 ppm and pressure between 315 and 420 psig.

At the end of this time period the unit experienced total shaft seizure probably resulting from the melted stator bore seal causing interference between the alternator rotor and stator which resulted in overloading of the alternator bearing. This shaft stoppage occurred even though full inlet pressure was being exerted on the turbine.

e. Three Minutes and Thirty Seconds

The turboalternator shaft seizure occurred at the start of this time period. At the end of the period it was realized that the turbine inlet air operated solenoid valve was disabled. The boiler discharge manual valve was then closed which immediately stopped vapor flow to the turbine.

With the shaft speed at zero rpm there was of course no package pump discharge pressure nor any alternator electrical output.

Even though 1250°F mercury vapor was still being supplied to the unit the alternator temperature was decreasing. The connector shell temperature dropped to 470°F and the alternator coolant outlet temperature decreased from 635°F to 590°F. This indicates that the alternator bore seal over temperature was not caused by the vapor flowing through the turbine during this time period. Therefore the bore seal failure must have been caused by the high temperature generated within the alternator which resulted from the short in the electrical connector.

After turbine vapor flow was ultimately terminated, lube flow was maintained to the bearings and coolant flow to the alternator coolant jacket. It was not apparent at that time that the alternator bearing had failed nor was it certain that the shaft had seized. The short in the electrical connector was discovered leading to the decision to completely shut down the test rig until an evaluation of the condition of the condition of the turboalternator could be made. After it was found that the bore seal had failed and the shaft was seized the unit was removed from the test rig.

5.0 FAILURE ANALYSIS

Previous sections of this report have discussed in some detail the sequence of events which occurred during failure of the unit. It has also been indicated that the short in the rig electrical connector initiated the failure and subsequent damage to the unit is consequential to the prime failure mode. Therefore the failure analysis will deal primarily with determining the cause of the short between phase one and neutral of the electrical connector.

During the final two hundred hours of operation the turboalternator had been operating with the alternator rotor partially demagnetized. The resultant higher current imposed a marginal operating temperature condition on the electrical connector. This was recognized at the time and after careful evaluation of the situation it was decided to continue the test. The connector temperature was monitored continuously and up to the time of the short did not exceed the accepted rating of 350°F.

To summarize the events leading up to the short the following outline is presented:

a. Connector History

The rig side electrical connector used for all Sunflower turboalternator testing was a standard Deutsch Co. female connector, part number DS07-197S. Early in November of 1961 it was soldered into the component rig wiring harness, the solder used being 95% tin and 5% antimony which has a melting point of 450 to 460°F. It successfully withstood a 1500 v.d.c. hi-pot test with no breakdown. The accepted rating of this connector is 40 amps per pin (there are 7 pins) at sea level, with an operating range of -67°F to 350°F. This connector was part of the test rig wiring from November of 1961 until September of 1963, an elapsed time of 22 months or nearly 16,000 hours. During this period it was cycled from room ambient to operating temperature six times during the various turboalternator tests.

Based on data obtained in the final hours of the CSU I-3A-2 Test it is estimated that during normal usage the connector temperature was in the 300 to 320°F range. Of the seven pins in the connector, current flows through only three with 14 to 15 amps being carried in phases one and two and 20 to 21 amps in the neutral. Two other pins were used for a thermocouple connection and the two remaining were unused. Thus the connector was actually operating in a conservative manner and could reasonably have been expected to last for the balance of the CSU I-3A-2 endurance run were it not for the unexpected chain of events during the final hours of operation.

b. Off-Design Connector Operation

All but about 300 of the 6741 operational hours were at the conditions indicated above or at less severe conditions. During the CSU I-3-1 test the turboalternator was operated at above normal power levels for a period of about 100 hours. The other 200 hours of off design connector operation occurred during the final hours of endurance testing of CSU I-3A when the alternator rotor was operating partially demagnetized which reduced the voltage and thus increased the current. The connector was then required to handle 21 to 22 amps in phases one and two and 30 to 31 amps in the neutral. A thermocouple added to the connector shell indicated a marginal but safe operating temperature of 330 to 350°F.

c. Final Temperature Cycle

The test shut down resulting from the municipal power failure was of sufficient duration to allow the connector to completely cool before the unit was restarted. Thus the connector cycled from 340°F to 75°F and back to 340°F. Condensation or thermal distortion caused by this cycle may have contributed to the connector failure.

The short in the connector was immediately apparent when the turboalternator was examined at the time of failure. The silicone rubber insulator in the connector was cracked and charred around the phase one and neutral pins. These two pins were connected to one another by the fusion products of the arcing. When the failed connector was removed from the alternator connector the two shorted pins remained connected to their mating pins on the alternator connector. A considerable force was required to remove these two pins and in the process they became separated. There was no apparent damage to the connector on the turboalternator except for some hairline cracks in the glass insert around the two pins that were shorted; but this undoubtedly was caused by the forceful removal of the mating pins.

Visual and metallurgical examination of the connector shortly after shut down suggested several possible causes of its failure which are as follows:

a. The connector shell temperature was at 350°F at the time of the short and it was logical to suspect that the pin temperature was even higher. There was therefore the possibility that the solder melting point of 450°F could have been reached allowing the melted solder to short the two pins.

b. Thermal distortion caused by the final temperature cycle could have shifted the two pins or lead wires close enough together to permit arcing.

c. Condensation resulting from the temperature cycle could have increased the contact resistance which in turn may have produced higher pin temperatures that could have melted the solder.

d. The final possibility was mechanical movement of the connector or its lead wires could have moved the two pins or their wires in close enough proximity to cause arcing. Deterioration of the silicone rubber insulator with time could possibly have permitted this to happen.

In hopes of pin-pointing the cause of failure a series of tests were performed on a rig side connector identical to the one which failed. An attempt was made to simulate actual operating conditions by using the same alternator connector that was on the turboalternator, heating it to the proper temperature, and passing the same current through it. The rig connector wires were soldered with the same type solder as used in the original rig connector. Thermocouples were imbedded in the solder at each pin connector to measure the solder temperatures under operating conditions. The alternator side connector was instrumented with a thermocouple to measure its shell temperature which was controlled by a heater wrapped around the connector.

Using the set-up just described the connector was operated at various power levels and connector shell temperatures; some, more severe than those existing at the time of the failure. The highest pin temperature reached during these tests was 289°F, far below the melting point of the solder. It was decided that these tests did not accurately simulate actual CSU test conditions so a new connector, identical to the one that was on CSU I-3A, was obtained.

The new CSU connector had a full lead tube connected to it so that it could be properly heated for more accurate simulation of actual test conditions. A series of tests was run with this new connector and this time the pin temperature climbed as high as 390°F but this was at a higher than normal power level and a connector shell temperature appreciably higher than normal. Analysis of the data indicated that, at the connector temperatures which existed at the time of failure, the average pin temperature was slightly over 290°F. A summary of the connector test results is given in Figure 17.

Since this pin temperature could not have melted the solder the other possible causes of failure were investigated. Contact resistance of the failed connector assembly and of the new connector assembly was measured and found to be quite low and of the same order of magnitude. Even deliberate attempts to raise the contact resistance of the new connector by damaging the pin surfaces did not cause it to increase. The assembly was retested to see if any of this had an effect on the pin temperature at operating conditions. The results were the same as for the previous test and the pin temperature of the "damaged" condition was not measurably different than the other two.

One other possible cause of failure, movement of the connector or its lead wires, was investigated. As previously indicated the connector was soldered with the same type solder as used on the failed connector. Also it was temperature cycled during this test series at least as many times as the failed connector and, except for during the CSU failure, was subjected to higher temperature. Therefore, aside from operational life considerations, the connector used in this test series had withstood treatment at least as rigorous as the failed connector. It still would not fail even when the lead wires were forcefully moved from side while it was carrying full current. However, one fact was brought out during metallurgical examination concerning the type solder

SUMMARY OF CSU CONNECTOR TEST RESULTS

Connector Assembly Tested	Run No.	Power Carried by Connector Average Per Pin			Connector Shell Temp. °F	Average Pin Temp. °F	Bellows Temp. °F	Heater Power Volt-Amps
		Volts	Amps	Watts				
Original Assembly	1	0	0	0	300	201	Not Available	34.7
	2	110	14	1335	335	243		34
		110	17	1705	350	261		35
		110	21.5	2130	372	289		34.7
	3	70	17.8	1150	304	229		36.2
		70	21	1400	319	251		36.2
New Assembly	1	70	15	965	359	286	615	67
		70	20.7	1335	390	332	640	67
		70	26	1725	430	388	671	67
	2	110	14	390	370	294	623	67
		108	20	1235	405	347	655	67
		108	25	2260	433	390	681	67
	3	70	13.9	720	305	243	420	80
		70	21.6	1290	332	286	440	82
Actual CSU Operating Data	1	70	24.3	1400	335	?	450	
		70	20	1130	300	?	450	

FIGURE 17

that was used. When the joint temperature exceeds 250°F, mutual diffusion of copper and solder results in the formation of brittle copper-tin compounds. The joint becomes brittle and may fail if it is subjected to movement. This condition becomes more severe with time and could not be simulated with the new connector.

Based on the results of the connector tests described above it would appear that the failed connector should not have shorted out as it did. But the fact that it did fail leads to the conclusion that the test series did not accurately simulate the actual turboalternator test conditions.

Because of the questionable areas regarding these connector tests it is impossible to pin point the cause of the connector failure with real certainty. More than likely it was a combination of things, such as a mechanical movement of the lead wires (which could have occurred during the shut down prior to last run) and then some thermal distortion occurring during reheat. This could have placed the two leads in close enough proximity to have permitted arcing, especially if the silicone rubber insulator had deteriorated with time. In any event precautions will be taken to prevent future recurrence of a connector failure.

6.0 CSU PACKAGE AND COMPONENT OPERATIONAL PERFORMANCE

The CSU I-3A package performance was good for the conditions of operation relative to its design specifications. However, several problems prevented complete attainment of design performance during the period of time when turbine deposits were not a factor. This section evaluates overall CSU performance as well as CSU subcomponent performance under conditions of operation. A further objective of this section is to point up necessary corrective action to eliminate problems and obtain design performance.

6.1 CSU Package Performance

Overall CSU package performance varied to a certain extent during the 4328 hours of operation mainly because of first stage nozzle blockage and rotor drag due to liquid mercury in the alternator rotor cavity. Therefore overall package performance will be evaluated for the period of 3000 to 3400 hours of CSU I-3A-2 test. During this period the cavity drain flow was not eliminated but was in general minimized and, based on post-test nozzle area measurements, the first stage nozzle was not restricted. Average output power was 2850 watts. The factors which controlled overall performance are listed below:

a) The turbine first stage nozzle throat area was measured by controlled nitrogen flow-pressure tests to be .0267 in^2 compared to the design throat area of .0293 in^2. This resulted in turbine vapor flow being 9% low at design nozzle inlet pressure.

b) The liquid screw seal between the alternator rotor and journal bearing permitted leakage of liquid mercury into the alternator rotor cavity. The effects of this leakage have been evaluated for this unit.

c) Turbine inlet temperature (at nozzle entrance) was deteriorated relative to the controlled 1250°F inlet stream because of excess heat loss from the 360° inlet collector scroll. The scroll exposed roughly eight to ten time the heat transfer area between turbine inlet flange and nozzle entrance compared to the original direct entry design. This not only reduced nozzle inlet enthalpy (ideal available power) but also is suspected of causing greater moisture drag losses, especially in the second and third stage wheels. This problem was at least partly alleviated however, by maintaining power to the turbine inlet housing heater.

During the time period selected to evaluate package performance, the following operating conditions existed:

Speed 40,000 rpm
Turbine Inlet Pressure 225 psig
Turbine Inlet Temp. 1250°F
Turbine Exhaust Pressure 7.0 psia
Turbine Exhaust Temp. 605°F
Alternator Cavity Drain Flow .06 ppm
Measured alternator output power 2850 watts (includes wattmeter correction)

To determine the alternator output power the turboalternator would produce if it weren't for items a, b, and c above, it is necessary to calculate the turbine output power. This was done using the following relationship:

$P_t = Pa\eta a + Ps$

where P_t = turbine output power.

Pa = alternator output power

ηa = Alternator efficiency including windage.

Ps = Fixed shaft losses (at constant speed) absorbed by bearings, pump, and other drag losses not associated with the turbine or alternator.

$\eta_a = 86\%$ (from component test data)

Bearing power is 600 watts total and pump power is 230 watts (from component test data). Shaft drag due to seal leakage into the alternator cavity is 235 watts (for zero leakage) from Figure 14. Therefore $P_s = 600 + 230 + 235 = 1065$ watts.

$$P_t = \frac{2850}{.86} + 1065 = 4385 \text{ watts}$$

Since the first stage nozzle is 9% undersize (item a) the turbine output power with the throat area corrected to design would be 4385/.91 = 4820 watts. Shaft drag due to seal leakage (item b) would be zero which reduces Ps to 830 watts. The turbine housing heater has minimized the effect of heat loss from the inlet scoll (item c). Therefore the projected output power of the turboalternator unit is as follows:

$$Pa = \eta a\ (Pt - Ps) = .86\ (4820 - 830)$$
$$Pa = 3430 \text{ watts projected output}$$

This value agrees reasonably well with the projected output of CSU I-3 which was 3560 watts and the projected output of CSU I-1A which was 3490 watts.

6.2 Turbine Performance

The throat areas of the nozzles used in CSU I-3A were all less than design; first stage 91%, second stage 94%, and the third stage 93.5% of design. Since all three were about the same relative to design values no correction concerning efficiency had to be applied because of this deviation. Thus the efficiency could be calculated directly from the basic relationship

$$Pa/\ \ a = \eta_t\ W\ \ H - P_s$$

where Pa = alternator output power = 2850 watts

ηa = alternator efficiency = 86%

ηt = turbine efficiency

W = mercury vapor weight flow

W = 91% of design = 12.45 ppm

H_{in} = inlet enthalpy = 161.7 btu/lb.

H_{out} = discharge enthalpy = 121.0 btu/lb.

H = 161.7 - 121.0 = 40.7 btu/lb.

P_s = 1065 watts

$$\eta_t = \frac{Pa/Na + Ps}{W \times H} = \frac{\frac{2850}{.86} + 1065}{12.45 \times 40.7} \times .057$$

$$\eta_t = 49.3\%$$

This value for turbine efficiency is slightly lower than the design value of 51% but this can be attributed to several factors. The assumptions used in the parasitic shaft loss (Ps) evaluation were all considered in a conservative manner. Furthermore, the heat transfer effect in the first stage inlet collector on both total ideal enthalpy drop and possible moisture drag in the turbine was not considered in the efficiency determination.

General conclusions from the CSU I-3A test related to turbine performance are:

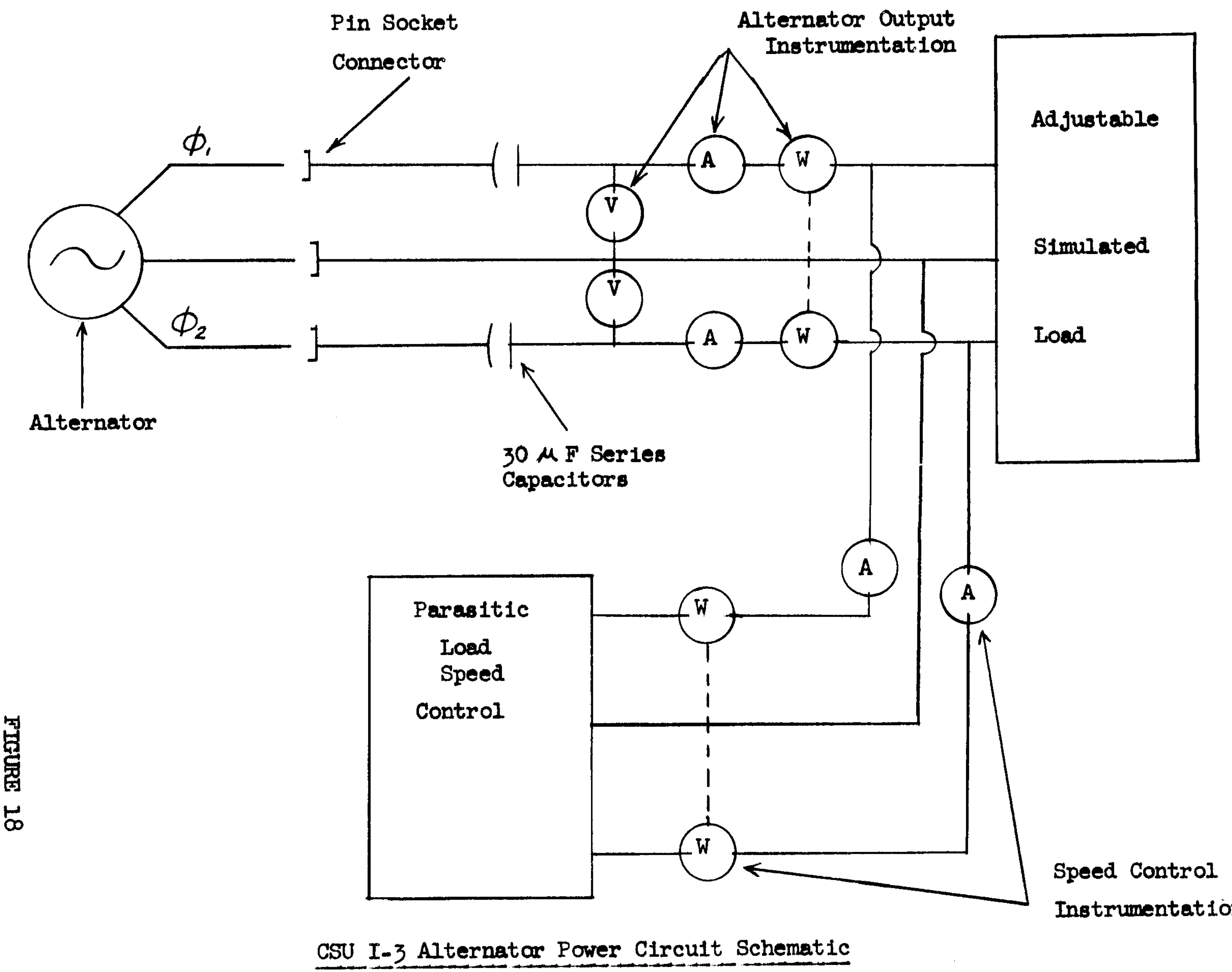
a) No basic aerodynamic redesign or parametric changes are required for the Sunflower turbine to meet its specified design performance of 51.0 percent efficiency and 5000 watts shaft output power.

b) To physically meet the design specifications, attention must be focused on fabricating nozzles to their throat areas.

c) The turbine inlet collector heat transfer problem must be corrected to avoid robbing the turbine of ideal energy and to reduce moisture content within the turbine. This latter item is important in reducing moisture drag on the turbine, but even more important in terms foreign deposits and turbine blade erosion.

6.3 Alternator Performance

Alternator electrical performance during the turboalternator test is best described in terms of its involvement with other components in the electrical circuit. Figure 18 is a schematic drawing of the test rig electrical network. The capacitors provide protections against short circuit



CSU I-3 Alternator Power Circuit Schematic

FIGURE 18

demagnetizing of the permanent magnet rotor which could cause loss of load carrying ability and subsequent unit overspeed. The parasitic load speed control senses alternator frequency and regulates its own load resistance such that the total load (including the simulated load) controls speed to a constant value. The simulated load is manually adjustable to obtain any desired fixed load in the range of operation. The following paragraphs generally describe electrical performance in terms of related circuit components.

6.3.1 General

The alternator electrical characteristics, as measure from subcomponent dynamometer tests are shown in Figure 19. This figure is typical of the engineering data developed for each alternator stator and is the same information used in the turbine efficiency calculations. During the CSU test at design speed of 40,000 rpm, the alternator produced various power outputs ranging from 1000 to 3000 watts at the capacitor output terminals, depending on the turbine operating point. The voltage output was initially 114 volts (phase 1) and 111 volts (phase 2) with respect to neutral. For most of the test, the average power factor was about 0.9 lagging for the two phases. A slight difference existed between phases but was probably due to frequency instrumentation and speed control sensing circuits which were all on the same phase. The lagging characteristic was a function of the combined circuit elements, including the alternator, since the simulated load was always kept resistive.

6.3.2 Alternator

The stator portion of the alternator was used in the CSU I-3 2350 hours endurance run and was reused in CSU I-3A without rework. It thus accumulated a total of 6676 hours of operation with all but the last 300 at, or very near, design level. As explained earlier in this report the permanent magnet rotor was inadvertently partially demagnetized to a level of 70 volts near the end of the CSU I-3A-2 test. With the turbine operating at design conditions the output power was reduced about 50 watts which indicates a decrease in efficiency from 86% to 84%.

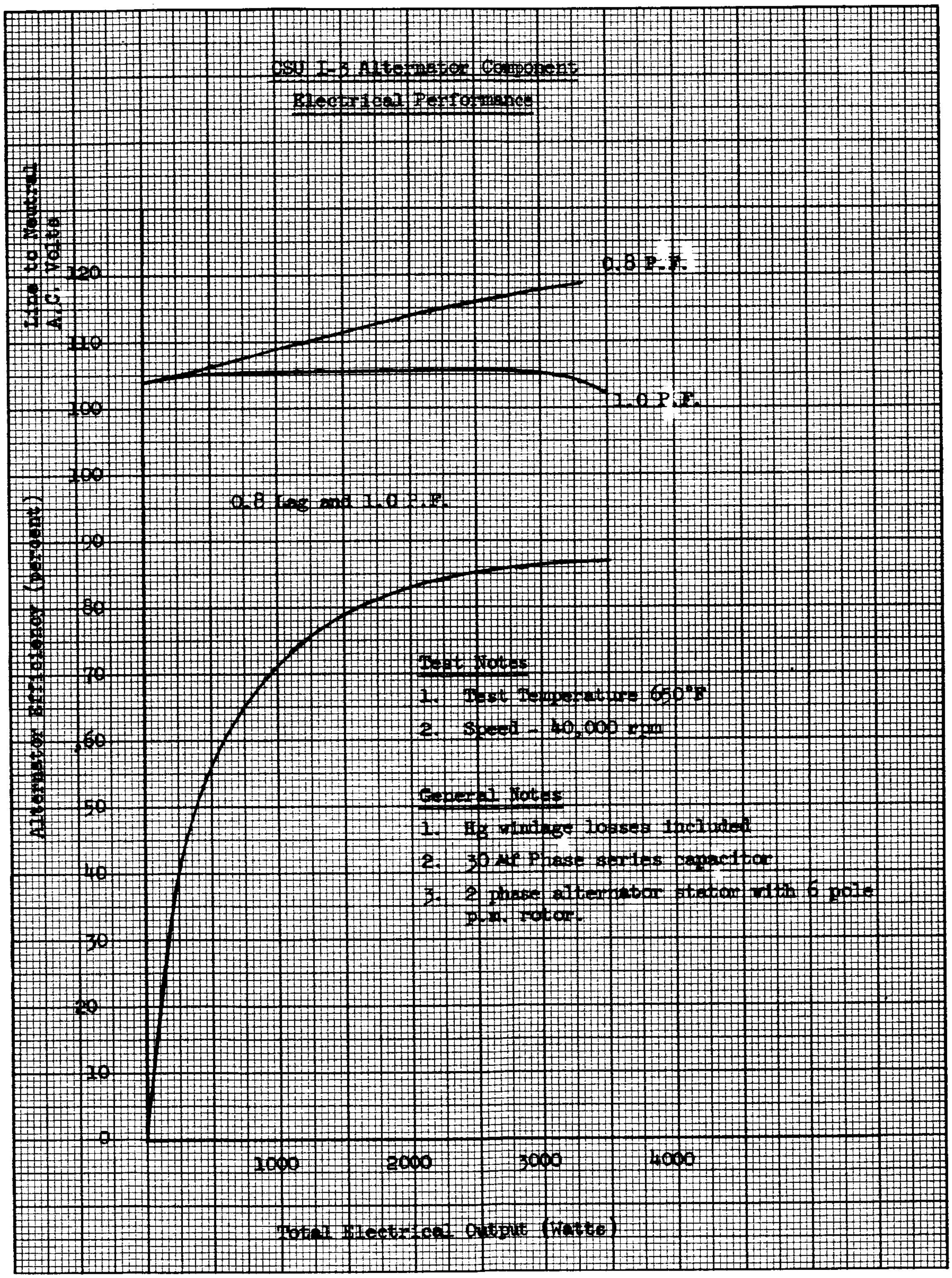

CSU 1-3 Alternator Component
Electrical Performance
Line to Neutral
A.C. Volts
120
110
100
0.8 P.F.
1.0 P.F.
100
90
80
70
60
50
40
30
20
10
0
Alternator Efficiency (percent)
0.8 Lag and 1.0 P.F.
Test Notes
1. Test Temperature 650°F
2. Speed - 40,000 rpm
General Notes
1. Hg windage losses included
2. 30 μf Phase series capacitor
3. 2 phase alternator stator with 6 pole p.m. rotor.
1000
2000
3000
4000
Total Electrical Output (Watts)

FIGURE 19

A slight bore seal leak developed during the CSU I-3 testing so the normal precaution of maintaining a positive internal pressure in the stator was taken during the CSU I-3A test. Up until the final failure, when the glass bore seal melted, there was no positive indication that the leak became more severe.

Post-test inspection and metallurgical analysis of the alternator rotor disclosed a growth at the center of .005 in. in diameter. Three cracks in the rotor magnet were also discovered. This growth and the cracks in the magnet have been attributed mainly to creep and partially to the overspeed just prior to shaft seizure. Alternator performance appeared to be unaffected by these occurrences.

6.3.3 Speed Controller

The speed control used during this test was a console model with circuitry similar to the more compact prototype controller. It performed excellently, as evidenced by the extremely steady speed indication ($\pm$ 1 cps in 2000) and no requirement for drift adjustment or any other adjustments or repairs throughout the test. On the other hand, the total electrical circuitry in this test displayed characteristics which may be of interest in the design considerations for prototype speed controllers of this type. First, in changing the simulated load at constant alternator power output, a change in load voltage and current can be observed which suggests that the speed control reactance may vary somewhat over or within its range of operation. The degree and extent of this effect has not been measured but data is obtainable if needed.

A second effect noted was a current instability of $\pm$ 2 to 3 amps, with a frequency of 5 to 10 cps for at least two distinct speed control power points. The instability in each cause was easily removed by adjusting the simulated load to either increase or decrease parasitic power very slightly. In other words, the speed control power ranges in which instability occurred were very narrow (less than ± 25 watts). The condition does not occur in alternator - speed control bench tests using a similar electrical network, either with or without the capacitors. However, the bench tests used a dynamometer drive rather than a mercury turbine. Also, instabilities do not occur when a calibration power supply output is fed into the controller in place of the CSU output, either with or without the capacitors. Again, the power supply did not simulate alternator reactance or shaft dynamics. Based on these observations, the instability apparently represents a resonance involving not only electrical characteristics of the circuit components but also mechanical (rotational) dynamics of the CSU shaft.

6.4 Mercury Pump Performance

The CSU I-3A pump hydraulic performance appeared very good at the outset of testing and did not change during the CSU I-3A-1 test. Shortly after the start of CSU I-3A-2 test a deterioration of approximately 7% of the outlet pressure at a given flow was noted. After operating at these conditions for about 300 hours the pressure gradually rose to its original level where it remained for the balance of the test.

There was not real explanation for this variation in performance aside from the possibility of partial blocking of the jet nozzle. When the unit was disassembled corrosion products were found to have partially closed the nozzle opening. These deposits were soft, however, and were very easily removed. Similar corrosion products were found in the jet nozzle inlet filter and these were also easily flushed out. A flow check of the filter indicated a negligible pressure drop at its design flow. It is possible that during operation the nozzle could have become partially blocked with subsequent clearing causing the pump performance variations. In an attempt to answer this question, future units have pressure taps downstream of the jet nozzle which will measure jet nozzle performance.

Pump performance calibrations early in the CSU I-3A-1 test indicated that the pump exceeded design specifications at its normal inlet pressure of 5.0 psia and that it would meet performance at the minimum inlet 3.5 psia. See Figure 13 for a typical calibration curve for this pump.

As described in Section 7.1, paragraph e., the impeller experienced erosion and cavitation damage on the face of the pump and on the low pressure side of the blades. However, in spite of the damage noted there was no performance degradation during the test run.

Summarizing pump performance:

a) CSU I-3A pump performance was apparently compromised at certain times by partial restriction of the jet nozzle or filter.

b) CSU I-3 pump performance was equal to original experimental performance for this pump design.

c) Minimum suction pressure at which design performance could be met was determined to be 3.5 psia as compared to a minimum -1 g specification of 3.4 psia.

Design changes incorporated in later units to improve performance and reduce erosion are as follows:

a) Modification of impeller blade profile to better match the ideal curvature, which is more complex than the single radius curvature used in CSU I-3A. This modification is aimed at reducing or eliminating cavitation erosion in the impeller face.

b) Increasing the number of impeller back vanes. This change will reduce back face leakage due to a better dynamic seal which will decrease total impeller flow for a given useful flow.

6.5 Bearing Performance

Bearing fits and clearances were established from thermal stress computer analyses of both the turbine and alternator bearing structures in combination with thermal maps and results of previous turboalternator tests. The basic desire was to machine bearing clearances such that operating (hot) clearances would be in the range of .0010 to .0012 inches diametral. It was expected that this operating clearance would eliminate half speed whirl and yet not be so small that alignment problems would result. Assembly techniques had progressed to the point where nearly perfect bearing alignment could be achieved.

Based on the results of the CSU I-3 test, the CSU I-3A alternator bearing was expected to increase .0002 to .0003 inches from room temperature machined clearance to hot operating clearance. It was therefore machined to a .0090 inch diametral clearance. Operating clearance during the major portion of CSU I-3A testing was .0011 to .00115 inches which was very nearly the expected value.

Turbine bearing clearance was expected to decrease .0002 to .0003 inches from static room temperature to hot operating conditions. The CSU I-3A turbine bearing clearance was therefore machined to .00125 to .00135 inches diametral. Hot operating clearance determinations were clouded by a film that formed on the bearing journal which made the clearance appear to be smaller than it actually was. However, hot operating clearance was initially .0011 inches and this value agreed with the expected value.

The thrust bearing total axial clearance and rotating annular orifice clearance were established from CSU I-3 testing. Both of these clearances were reduced in CSU I-3A to lower the thrust bearing flow required to maintain adequate pressure. Axial clearance was reduced from .00105 to .00085 inches but this did not significantly change the bearing's flow pressure relationship. CSU I-3 thrust bearing flow was 15 ppm at 140 psig while for CSU I-3A it was 12 ppm at 105 psig. The same thrust bearing was used in both units and in both cases bearing performance was excellent. As previously mentioned this bearing was in excellent condition at disassembly after a total operating time of 6676 hours.

Half speed whirl was present for approximately the first 100 hours of both the CSU I-3A-1 and CSU I-3A-2 tests. During these stabilization periods the diametral clearances of both journal were somewhat higher than their nominal values during the tests. Once the clearances stablized there was no evidence of bearing whirl.

Turbine bearing clearance had a continual tendency to decrease throughout the two long CSU I-3A test runs. When the unit was parially disassembled after the CSU I-3A-1 test run a black powdery film was found on the journal. This film was easily removed and the measured clearance was the same as at assembly. At final disassembly this film was again present and again was easily removed. The final measured diametral clearance was identical to the original machined clearance after 4328 hours of operation. This film did not affect the operation of the bearing aside from causing an apparent decrease in diametral clearance. A constant lube flow was maintained throughout the tests and the condition of the turbine bearing was excellent at disassembly so there is every indication that the bearing would have survived the test objective of one years operation. See section 9.2 for a metallurgical analysis of the film on the bearing.

Bearing performance conclusions are as follows:

a) All three bearings operated satisfactorily throughout the CSU I-3A testing despite close operating clearances and the presence of significant shaft fundamental vibration.

b) The reduced journal bearing diametral clearance established for this unit successfully eliminated half speed whirl without introducing bearing alignment problems.

c) Thrust bearing load performance was more than adequate with 100 psig lube supply pressure.

No basic bearing design changes or bearing parametric changes are presently planned for either the journals or thrust bearing. However, there are existing modifications to the three sector bearing design which would provide whirl stability at larger operating clearances. Larger clearances are beneficial relative to bearing alignment and lube filtration requirements. However, the influence of modifications on load carrying capacity and sensitivity to other factors would need further evaluation. Such an evaluation is recommended either on a subcomponent basis or as an integral part of a CSU test to provide greater depth in bearing technology and application.

6.6 Screw Seal

A screw pump seal is used between the alternator journal bearing and the alternator rotor to prevent entry of lube drain liquid into the alternator rotor cavity. Indications are that this seal works at nominal bearing drain pressures for speeds up to about 30,000 rpm. But, above 30,000 rpm and specifically at the design speed of 40,000 rpm, the seal allows liquid Hg to leak into the rotor cavity. This leakage has several detrimental effects:

a) It consumes excess shaft power not only by drag on the alternator rotor but also within the seal itself because of an improper liquid-vapor interface.

b) It removes heat from the alternator which contributes to adverse alternator axial temperature gradients.

c) It causes rotor erosion and may have been responsible for a small crack in the alternator stator glass bore seal.

d) It necessarily passes out the alternator cavity drain (which was provided for nominal rotor cavity condensation) and into the turbine exhaust. An excess of liquid flow into the exhaust could upset or compromise balance and operation of the prototype condenser. In the CSU I-3A test, because of the drain flow measurement arrangement, this flow did not drain into the turbine exhaust line.

In addition to the above problems, the screw pump seal has been found to leak seriously (over 4 lb/min) when maximum drain pressures were imposed. The equivalent alternator output power loss between nominal exhaust pressure referenced drains and a 6 psi elevation in drain pressure was nearly 500 watts because of drag in the screw seal and on the rotor. The above observations have lead to a review of the seal requirements versus operational performance as related to its design. As a result, a new seal has been designed and found to meet the requirements on the basis of subcomponent testing. The new seal incorporates a multi-lead thread in the I.D. of the stationary member, with a smooth cylindrical mating section on the shaft. The original seal used a single lead thread (in CSU I-3A) on the shaft with a smooth cylindrical stationary member.

7.0 POST TEST TURBOALTERNATOR HARDWARE CONDITION

Shortly after the failure of the turboalternator it was removed from the test rig for disassembly purposes. Prior to removal, however, two preliminary investigations were made. The access plug in the turbine end of the unit was removed so that an attempt could be made to rotate the shaft manually. Application of a considerable amount of torque failed to move it. Also a pressure bleed down test was made on the alternator stator to determine the condition of the alternator bore seal. It was found to have a severe leak.

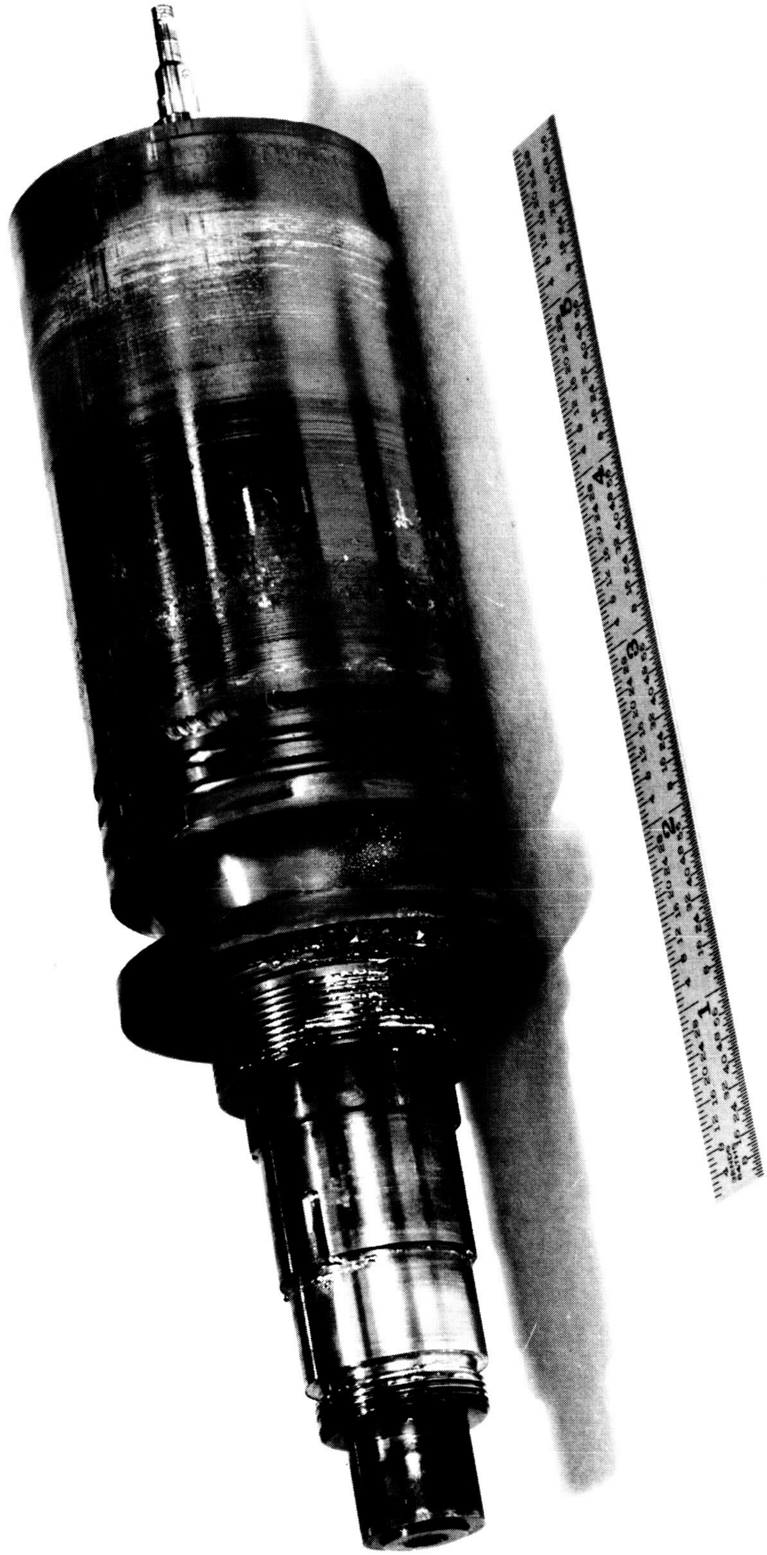
The following is a description of the various turboalternator components as they were found at disassembly of the unit.

7.1 Shaft

The shaft was bound up in the unit during the disassembly even after removal of all turbine wheels and nozzles. A straight applied force on the turbine locking nut was successful in extracting the shaft from the unit. Fig. 20 shows unit as removed in disassembly. It was then apparent that the shaft had contacted the alternator bore seal during rotation and that upon seizure of the shaft the alternator bearing journal had welded itself to the bearing bushing. The diameter on which the journal sleeve had been shrunk did not show that any relative motion had occurred between the shaft and the sleeve. An axial crack could be seen in the bearing sleeve. To remove this sleeve it was necessary to elox it in three places so it would collapse. Descriptions of the various shaft components are as follows:

a. Alternator Bearing

It would be difficult to assess the condition of the alternator bearing just prior to the CSU failure because of the galling and welding resulting from the seizure. However the thrust face of the journal sleeve is in excellent condition. Metallurgical examination of the crack in the sleeve indicates that the crack does not appear to be associated with the seizure of the bearing. The bushing had over nine months of operating time. See Fig. 21 for a photograph of the sleeve after removal from the bushing.



SUNFLOWER CSU I-3A SHAFT, PARTIAL DISASSEMBLY, PRIOR TO CLEANING AFTER 4328 HOURS OPERATION

FIGURE 20

SUNFLOWER CSU I-3A ALTERNATOR BEARING SLEEVE AFTER 4328 HOURS OPERATION

FIGURE 21

b. Turbine Bearing

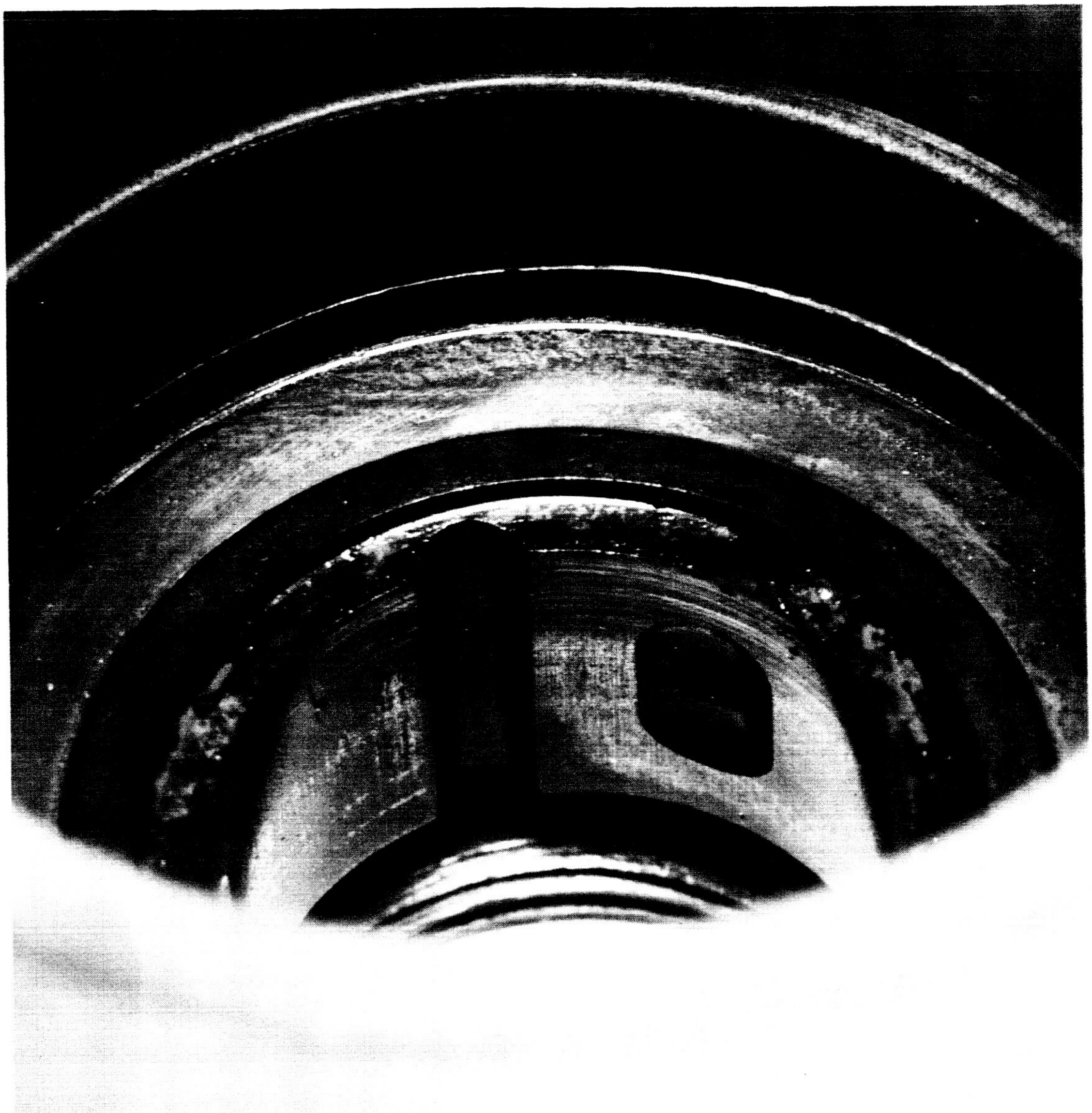
The turbine bearing bushing and journal sleeve are both in very good condition. Both surfaces are darkened in color with a thin film evident. This film is mainly iron with appreciable amounts of chromium and tungsten. There is one area on the turbine side of the bearing sleeve which appears to be a slight rub or burnish. It is about 1/8" wide and extends about 1/4 of the way around and most likely occurred during the final revolution. Similar markings are evident in the bushing. Dimensional checks of the bearing indicate that no wear has taken place in approximately 6 months of operation. The bearing sleeve can be seen in the photograph of the shaft (Fig. 20) and the bushing (Fig. 22).

c. Thrust Bearing

The thrust bearing stationary member (Fig. 23) was found to be in excellent condition. No dimensional changes were noted on this component which had operated for over nine months. A small amount of corrosion product (approximately .0002 in.) collected in the spiral grooves but this accumulation did not affect the performance of this bearing. The surface of the thrust washer was in excellent condition. This component had also accumulated over nine months of operating time.

d. Alternator Rotor

Melted glass from the stator bore seal was distributed over most of the rotor shrink can. This melted glass caused rubbing between the rotor and stator which led to the shaft seizure. The rubbing appears to have taken place over approximately 1/3 of the surface area closest to the alternator bearing (Fig. 20). Dimensionally the alternator rotor shrink can grew .001 in. at either end and .005 in. diametrally in the center. The actual pre-test clearance indicates .012 in. diametral clearance was available before contact would be made with the alternator stator. Removal of the shrink can disclosed three axial cracks in the rotor magnet, two of which ran the length of the magnet and the third one at least 1/3 of the length. Radial cracks were also visible in the center



SUNFLOWER CSU I-3A TURBINE BEARING BUSHING AFTER 4328 HOURS OPERATION

FIGURE 22

SUNFLOWER CSU I-3A THRUST BEARING AFTER 6676 HOURS OPERATION

FIGURE 23

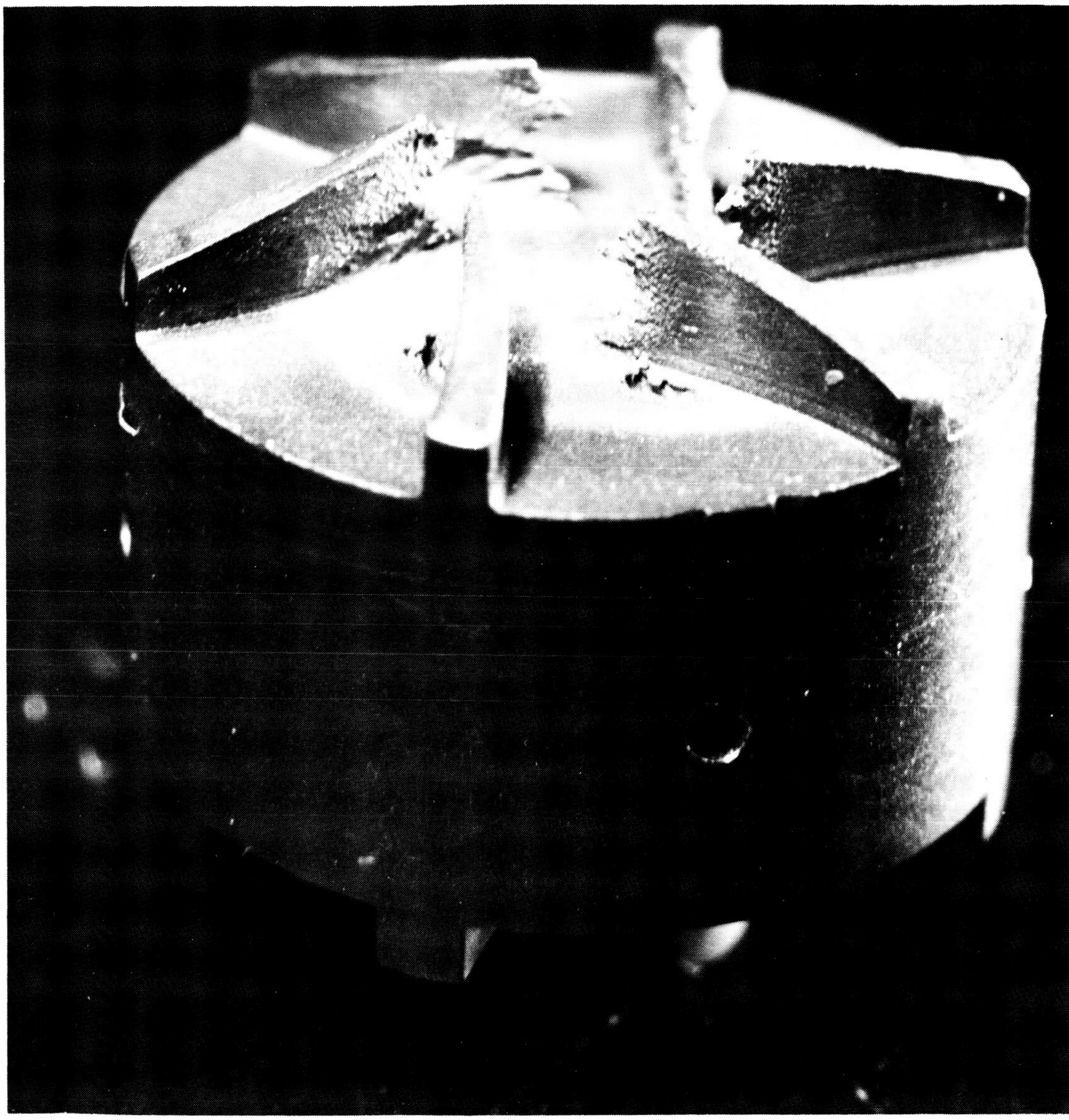
section of the magnet. The magnet has assumed approximately the same shape as the shrink can, i.e. it grew several thousandths in diameter at the center.

e. Mercury Pump

The mercury pump impeller shows similar type of erosion and cavitation damage as was experienced in the 2350 hour run in 1962. There was implosion damage on the high pressure side of the blade immediately after the flow is turned and directed along the blade contour. Apparently, formations of low pressure areas are prevalent in this area. The damage appears on the face of the pump, not on the blades. However, erosion is noted on the low pressure side or back side of the blade along a length approximately 1/4 of the blade length. Penetrations of approximately 4 to 5 mils are evident. Also, the eye of the impeller has experienced severe metal removal from all blades, the result being "jagged" edges on the entrance of the vane. Metal removal is in the order of .025". Figure 24 is a photograph of the impeller showing the damage described. This particular pump is the single radius blade curvature with a considerable mismatch of theoretical entrance angle due to fabrication compromises. However, in spite of the damage noted there was no performance degradation during the test run.

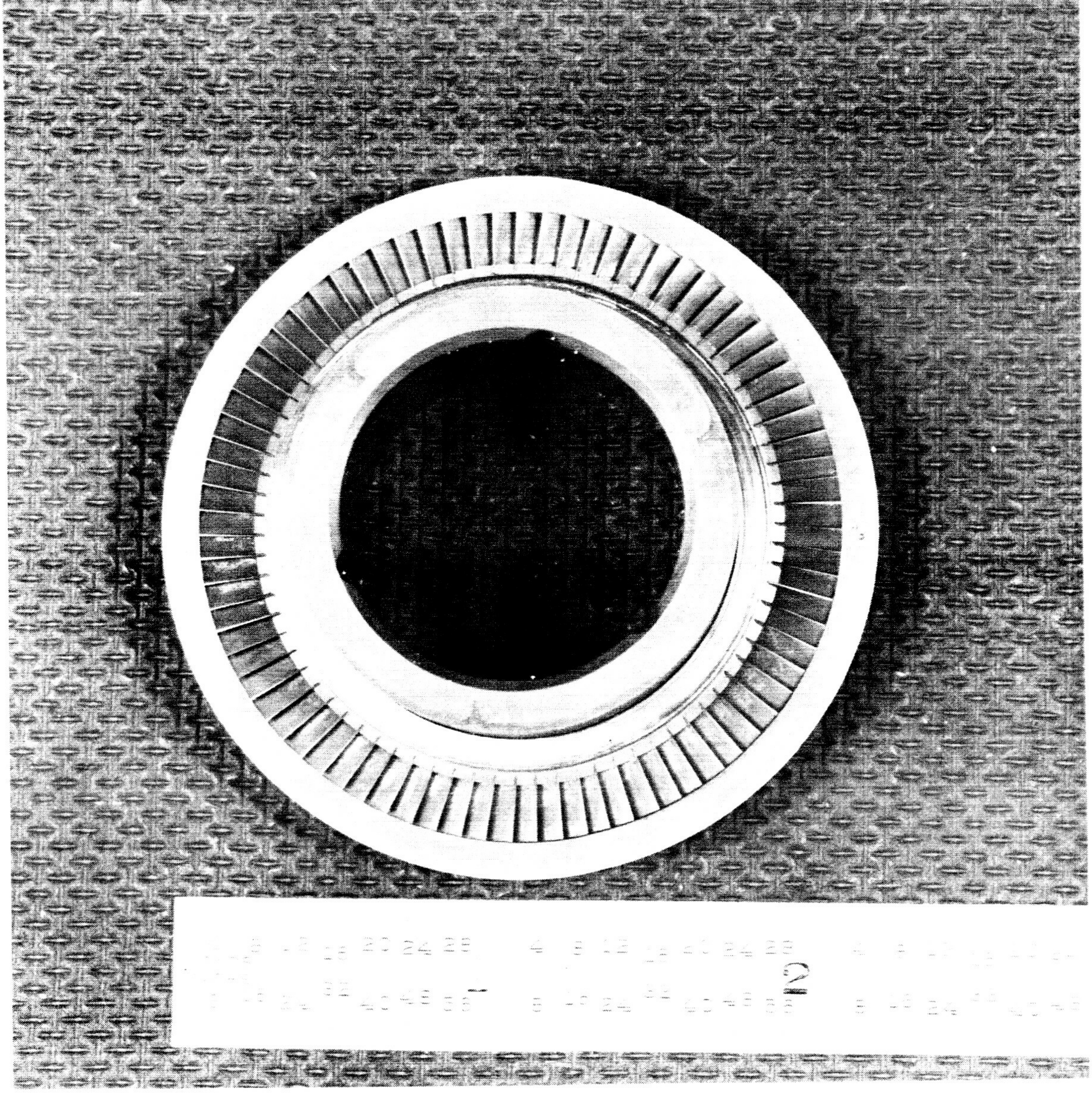
f. First Stage Turbine Wheel

The first stage turbine wheel (Fig. 25) sustained a significant amount of damage. During the final, or possibly the final few revolutions of the unit, contact was made between the turbine wheel intermediate ring and the second stage nozzle resulting in discoloration and metal galling. This also caused the intermediate ring, along with the blades and O.D. shroud ring, to be displaced approximately 1/32 of an inch toward the second stage wheel. The shroud ring of this wheel was loose with inspection indicating a diametral growth of .004 inches from the cold pretest measurement. The first stage turbine blading appears to be in good condition with negligible erosion damage present.



SUNFLOWER CSU I-3A MERCURY PUMP IMPELLER AFTER 4328 HOURS OPERATION

FIGURE 24



SUNFLOWER CSU I-3A FIRST STAGE TURBINE WHEEL DISCHARGE SIDE AFTER 4328 HOURS OPERATION

FIGURE 25

g. Second Stage Turbine Wheel

The second stage turbine wheel is generally in good condition. However, there was an increase of .0006 to .0014 inches in outside diameter and some slight erosion of the trailing edtes (Fig. 26). Also some slight deposition was noted throughout the wheel.

h. Third Stage Turbine Wheel

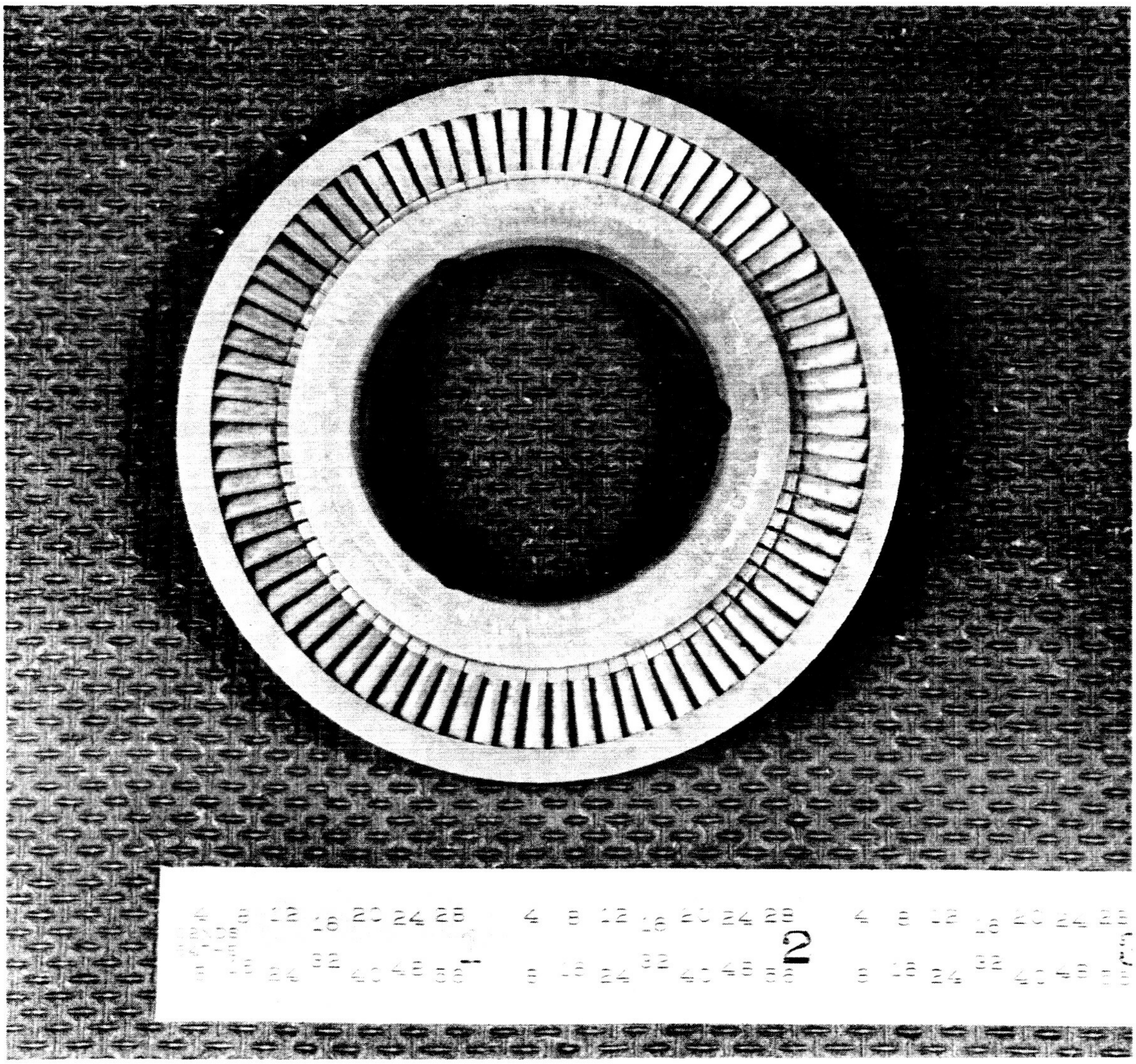
The third stage wheel shows slight erosion on the leading edge. The erosion is more pronounced than on either of the first two stages as would be expected from the higher moisture content at this point in the turbine. Also, discrete particles are observed to be missing on a few blades as if struck by a hard surface or mercury droplets causing a "chunk" to be dislodged from inlet side. This can be seen in Fig. 27. The particles might possibly be removal of ferritic stringers present in the material. This particular turbine wheel shows less erosion during its 6 months of operation than did the previous turbine after 2348 hours of exposure.

7.2 Turbine Inlet Housing

The turbine inlet housing appears to be in good condition. A small amount of corrosion product was present on the vanes of the first stage nozzle and on the upper and lower faces of the housing as shown in Fig. 28. However, no corrosion products of the type encountered in the 2348 hour run were found. In fact post-test cold gas turbine nozzle tests indicate that no change in first stage nozzle area occurred during the 4325 hours of testing. The final area was the same as the original measurement. The fact that this nozzle has remained clean for approximately twice as long in the same test rig can be attributed to two factors:

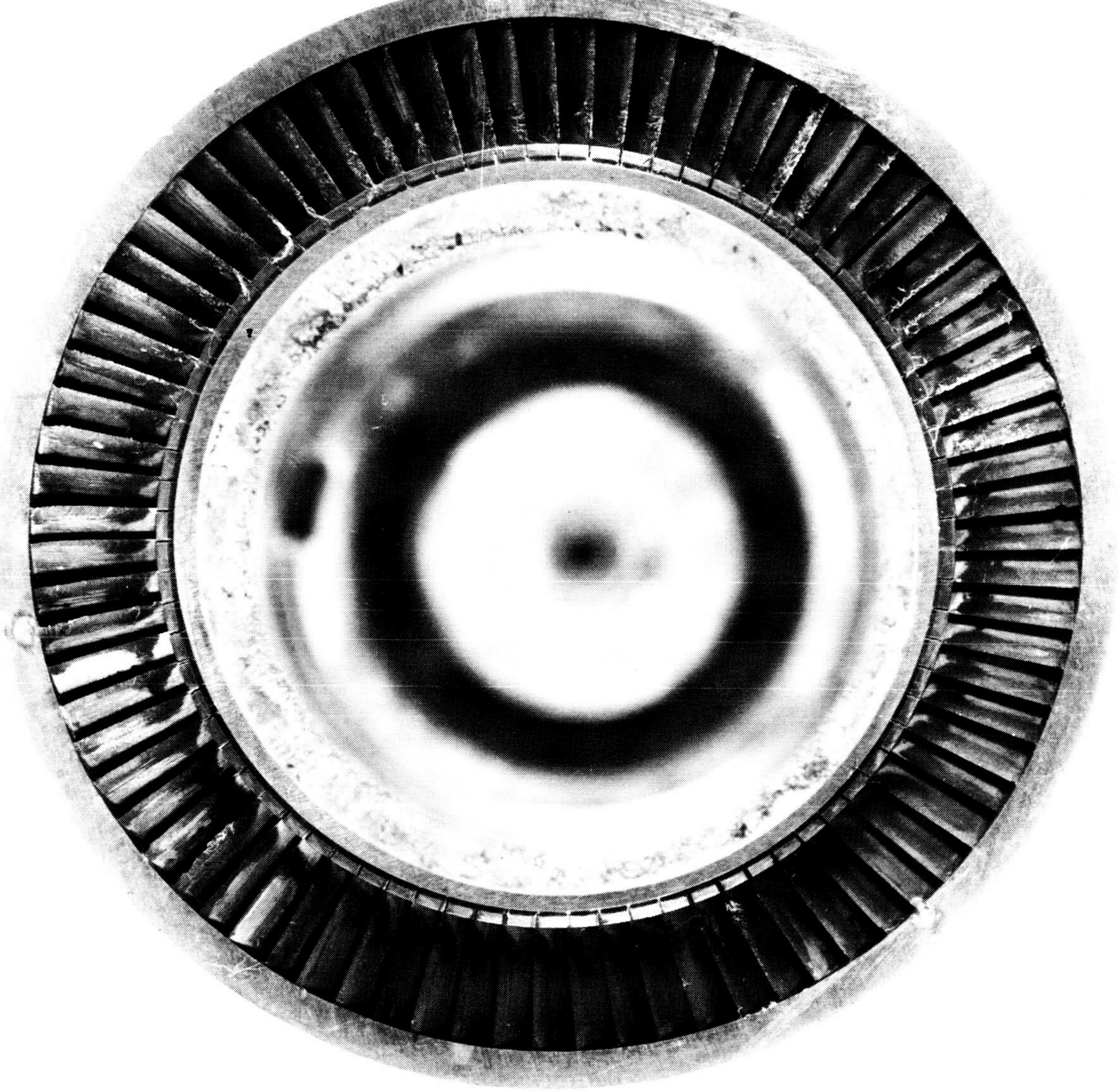
a. The maintenance of 50 watts of heater power on the 360° inlet scroll which reduced condensation and thus corrosion in that area.
b. Modifications made in the test rig vapor section.

Some erosion can be seen in the housing above the bearing.



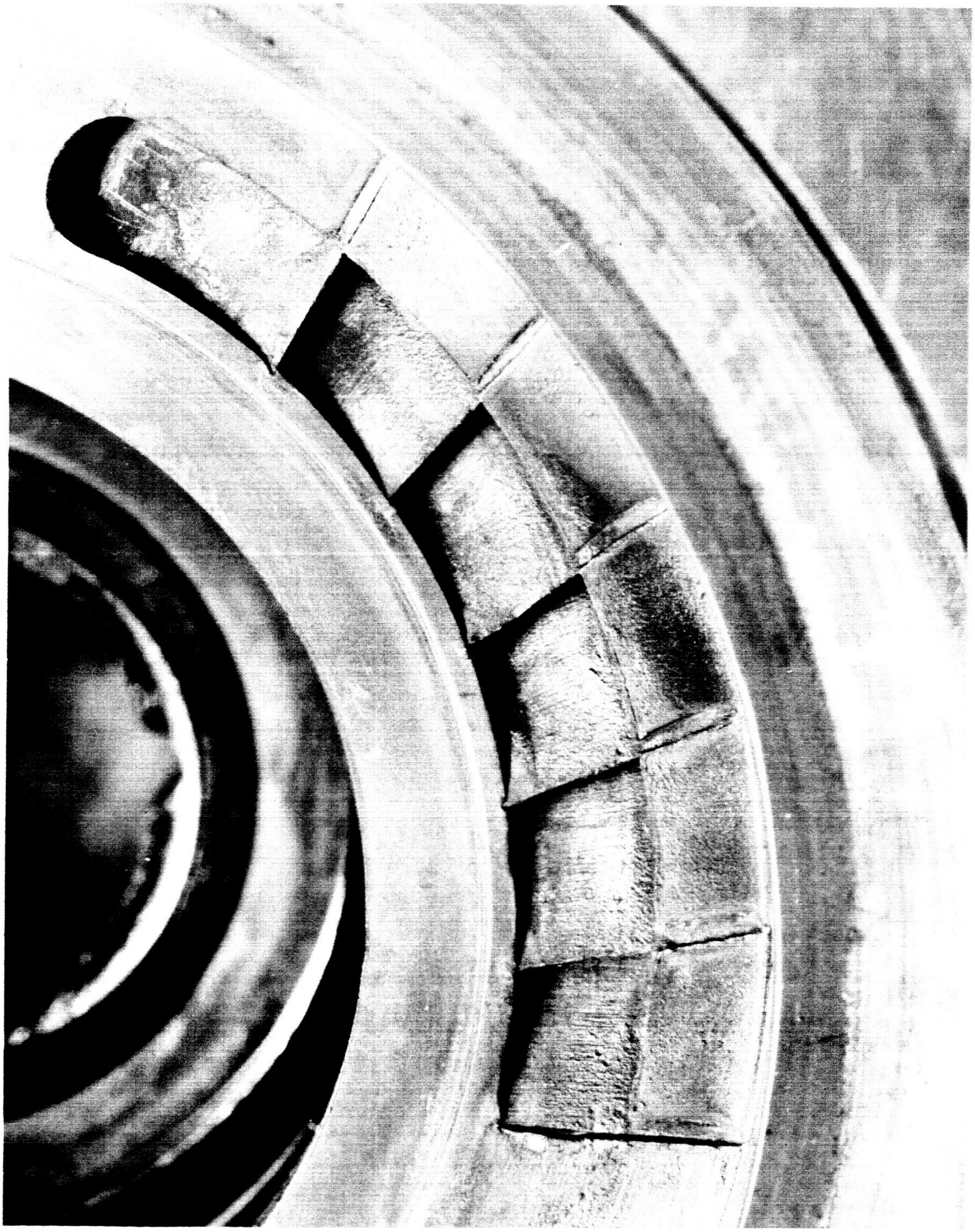
SUNFLOWER CSU I-3A FIRST STAGE TURBINE NOZZLE DISCHARGE SIDE AFTER 4328 HOURS OPERATION

FIGURE 26



SUNFLOWER CSU I-3A THIRD STAGE TURBINE WHEEL INLET SIDE AFTER 4328 HOURS OPERATION

FIGURE 27



SUNFLOWER CSU I-3A FIRST STAGE TURBINE NOZZLE DISCHARGE SIDE AFTER 4328 HOURS OPERATION

FIGURE 28

7.3 Second Stage Nozzle

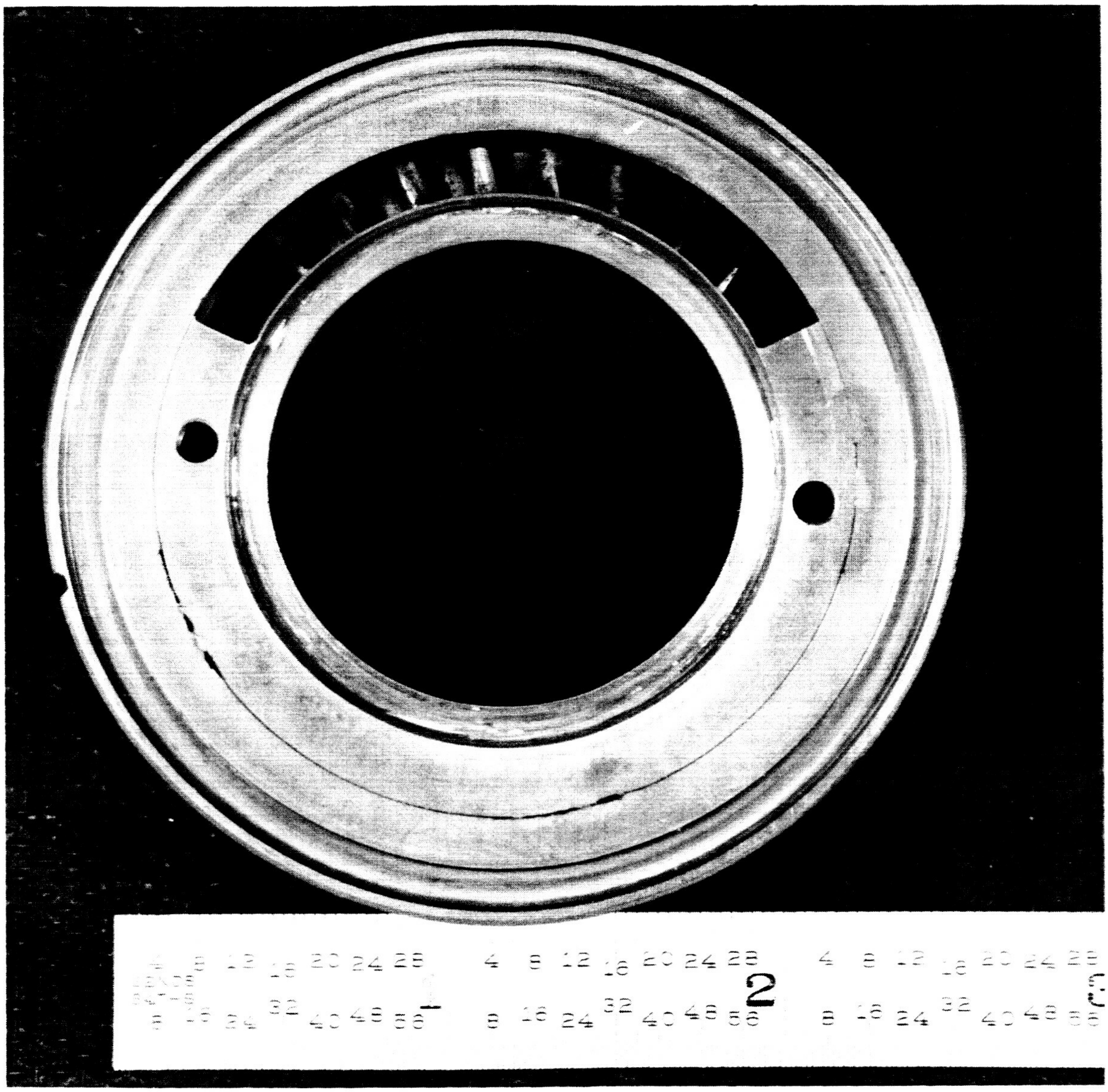
The second stage nozzle sustained some damage, the most notable being the result of the contact between the first stage wheel and this nozzle. Dimensional inspection indicates an out of squareness of .012 inches between the inner ring and the outside diameter. This cocking of the inner ring is what has the appearance of a crack in Fig. 29 . The inner ring was discolored and the surface galled when contact was made, probably during the last few revolutions. Two erosion grooves are evident in the inside diameter, one near the edge and one near the blades. The spacing of these grooves is consistent with the width of the second stage wheel. Apparently liquid was present at that location and was centrifugally thrown against the nozzle causing the erosion noted. Cold gas nozzle tests appear to indicate a 4 to 5 percent reduction in throat area. Deposits on the blades which are evident microsopically could account for this.

7.4 Third Stage Nozzle

The third stage nozzle is in good condition except for the same type of erosion grooves in the inside diameter as were found in the second stage nozzle. Also some deposit is clearly evident at the bottom in the direction of gravity indicating perhaps a floating of corrosion product on liquid which remained trapped in that area (see photograph Fig. 30). This would also explain a 4 to 5 percent reduction in throat area indicated by the post-test cold gas nozzle tests. Six of the original passages in this nozzle were blocked by pins to obtain the proper throat area. No substantial deposits appear to be collected in the blind passages around these pins.

7.5 Alternator Stator

The alternator stator was removed with ease from the alternator housing and sustained no damage aside from the melted glass bore seal. It was this melted glass which caused interference between the rotor and stator and ultimate shaft seizure. The turbine and portion of the stator appears to be "bubbly" in nature and pock marked. Figure 31

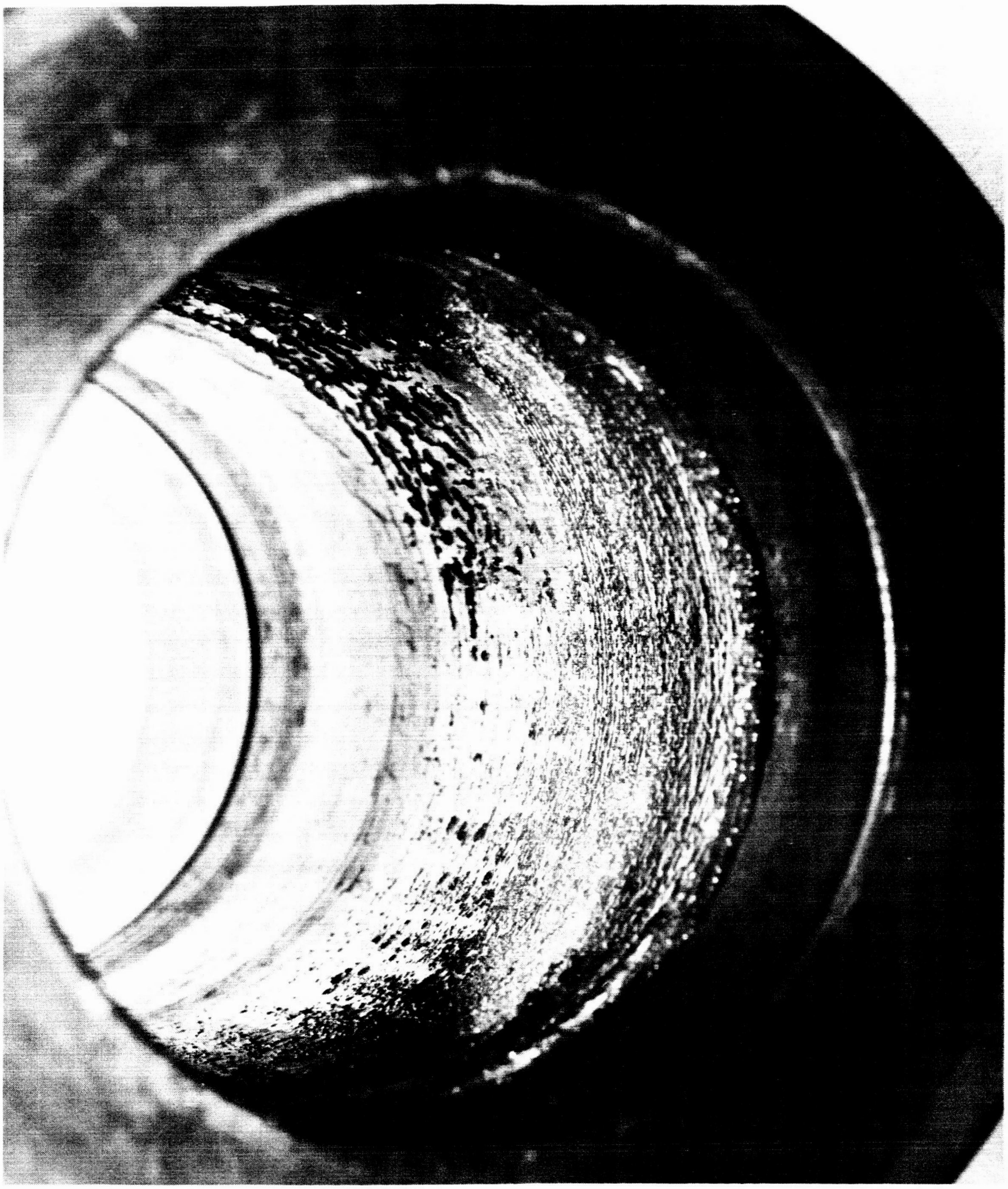


SUNFLOWER CSU I-3A SECOND STAGE TURBINE NOZZLE INLET SIDE AFTER 4328 HOURS OPERATION

FIGURE 29

SUNFLOWER CSU I-3A THIRD STAGE TURBINE NOZZLE DISCHARGE SIDE AFTER 4328 HOURS OPERATION

FIGURE 30



SUNFLOWER CSU I-3A ALTERNATOR STATOR BORE AFTER 6676 HOURS OPERATION

FIGURE 31

shows the stator I.D. Resistance checks of the windings seem to indicate that, electrically, the stator is in good condition.

7.6 Housings

Generally the housings are in very good condition having sustained no damage resulting from the unit failure. All except the turbine inlet housing have over nine months of accumulated operating time. Dimensional inspection disclosed no appreciable changes in diameters, parallelism, squareness, concentricities, etc.

8.0 TEST RIG TURBOALTERNATOR ELECTRICAL CONNECTOR POST-TEST CONDITION

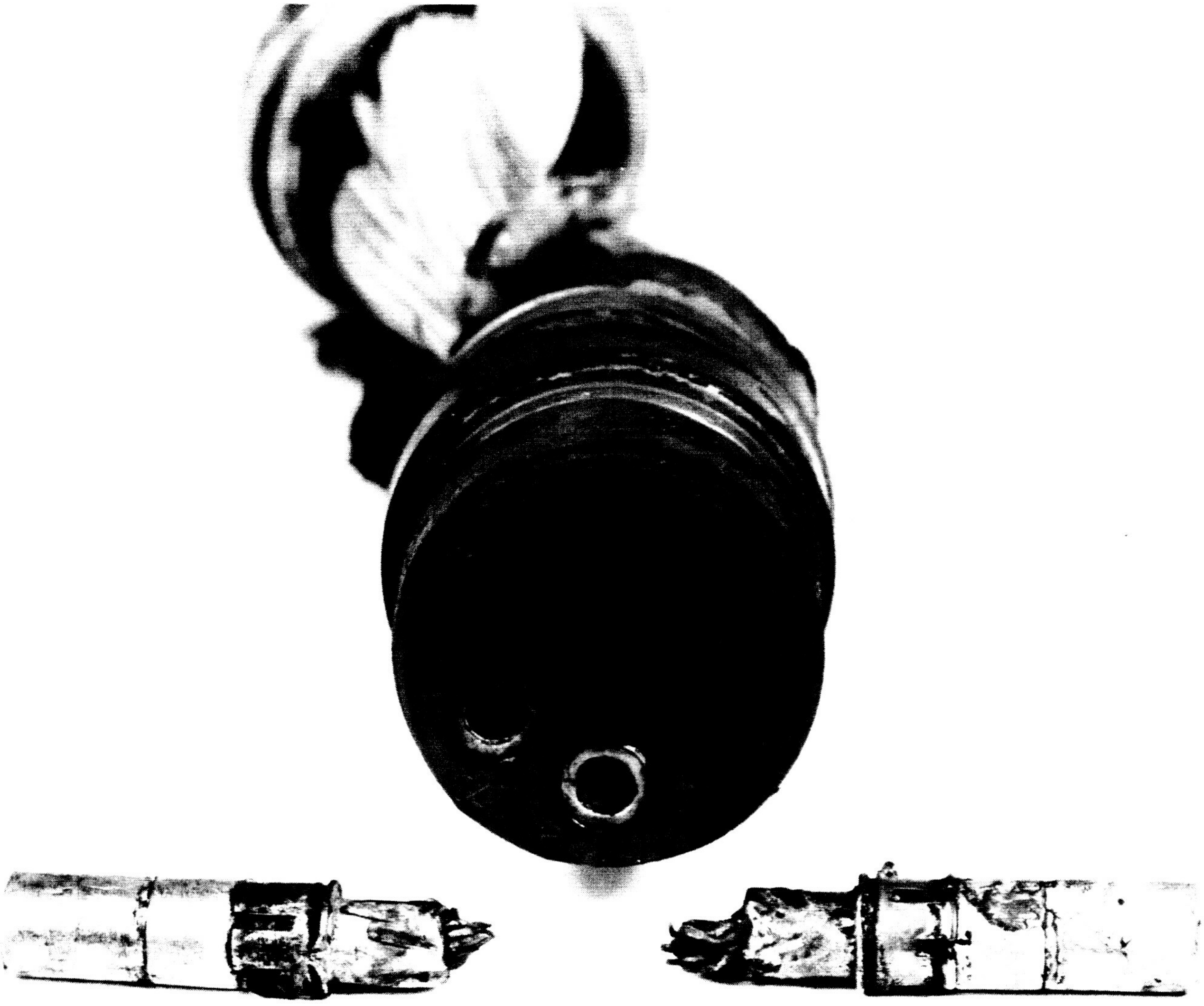
Examination of the turboalternator unit at the time of failure immediately disclosed the short in the test rig electrical connector. The silicone rubber insulator in the connector was cracked and charred in appearance. Closer examination revealed the short between the phase one and neutral pins. The two shorted pins were connected to one another by the fusion products of the arcing. Considerable force was required to remove them from the pins of the mating connector. Figures 32 and 33 are photographs of the failed connector.

The direct short through the contacts resulted in high temperatures being generated in the connector which were evidenced by the temperature logger traces which showed a rise from 350°F to 530°F in less than two minutes. This temperature was measured on the external barrel and is not a true measure of the temperature of the connector pins. In any event these temperatures are high enough to exceed the 450°F melting point of the solder used to make the electrical connections. It is uncertain whether the solder melted causing the short or whether something else caused the short and the resulting high temperatures melted the solder.

Microscopic examination of the connector revealed the following:

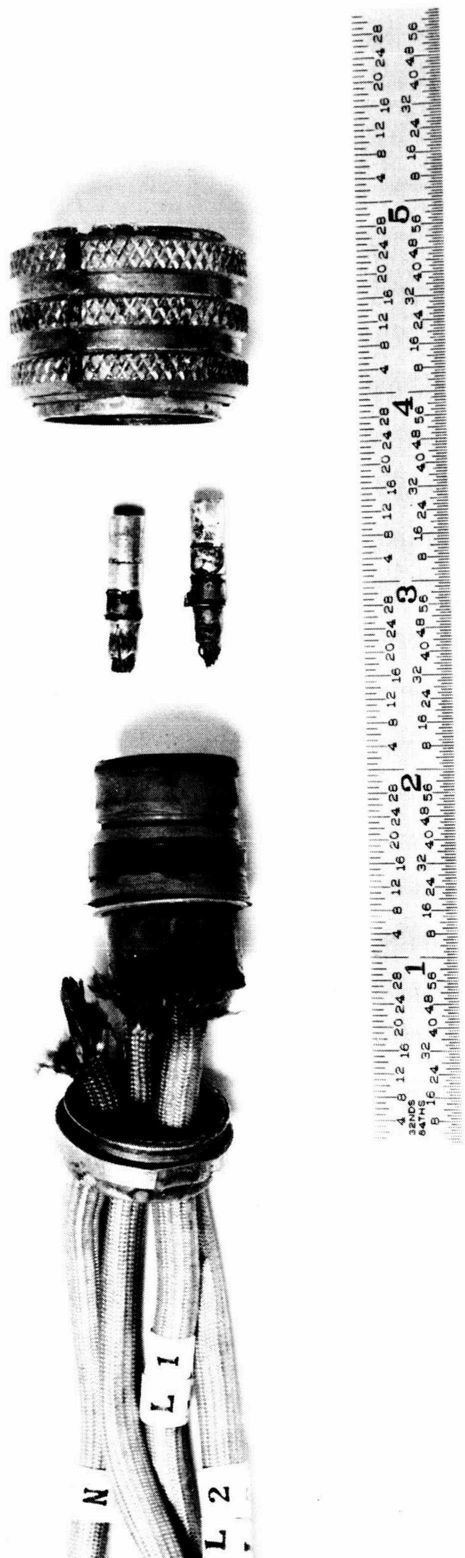
a. The solder melted and partially ran out.

b. The connector pins and copper wire strands were oxidized.



SUNFLOWER COMPONENT TEST RIG ELECTRICAL CONNECTOR AFTER 6741 HOURS OPERATION

FIGURE 32



SUNFLOWER COMPONENT TEST RIG ELECTRICAL CONNECTOR AFTER 6741 HOURS OPERATION

FIGURE 33

c. Some plating remained on the pin surface.

A check of contact resistance was made on one of the shorted pins. The value obtained was not appreciably different from that of a brand new connector which was also checked. A series of tests were performed on this new connector as well as the old one in hopes of determining the cause of failure. These tests and their results are discussed in section 5.0 of this report.

MATERIAL ANALYSIS

CSU turboalternator CSU I-3A components were submitted for microscopic scrutiny and analysis after exposure to operation of a mercury Rankine cycle system for periods up to 9 months in duration. A detailed analysis has been reported in Reference 1. The following paragraphs summarize the results of these evaluations.

9.1 Turbine Inlet Housing Assembly

A very small coating of corrosion product was noted on the PH 15-7 Mo vanes. Additional flow checks of the nozzle assembly failed to show any reduction in the area of the throat suggesting an absence of deposit in the metering area. The results are quite significant in that the unit operated twice the operating time of the previous unit without showing any of the same effects of nozzle blockage.

9.2 Shaft Assembly

The turbine bearing sleeve contained an extremely thin black film which was easily removed by scraping. The analysis of the deposit was as follows:

Major - Fe
Appreciable - Cr, W
Minor - Si, Mr, Mo, Ni
Trace - Cu

The constituents suggest dissolution of the bearing material itself. Microscopic examination showed localized corrosion to a maximum of .46 mils.

The turbine blades of the third stage wheel had evidence of erosion on the convex side of the leading edge, with little or no erosion on the blade surface areas. Erosion depth was estimated to be 1.6 mils. Some corrosion up to .4 mils was noted on localized working surfaces of the blades. An operating temperature of 700°F was confirmed from the hardness changes incurred during operation. The Larson-Miller master aging curve shown in Figure 34 illustrates how temperature predictions can be made.

The alternator rotor can showed no defects other than several pits and a growth change of .005 inches in the middle of the can which was attributed

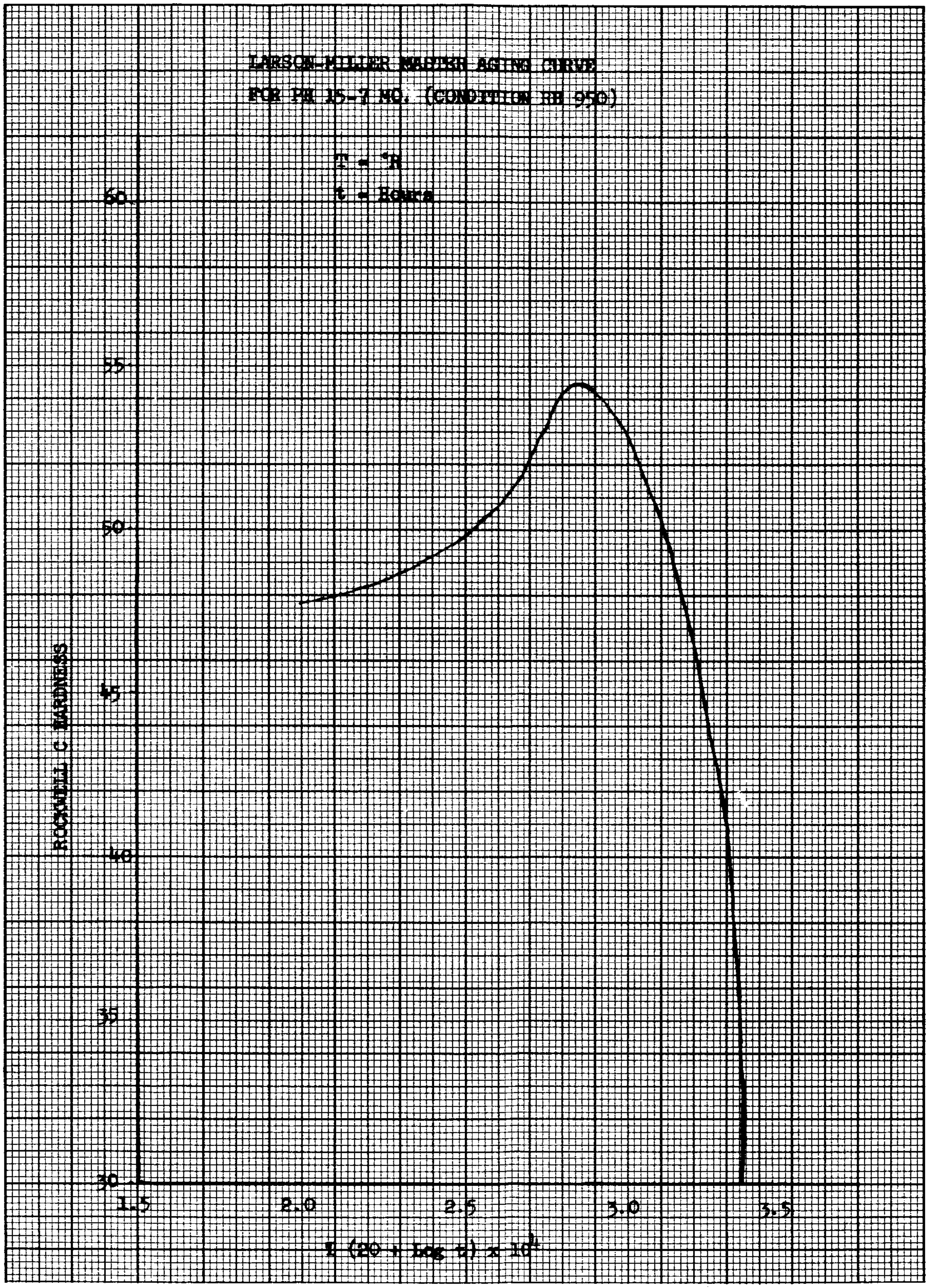

LARSON-MILLER MASTER AGING CURVE
FOR PH 15-7 MO. (CONDITION RH 950)
T = °R
t = Hours
ROCKWELL C HARDNESS
60
55
50
45
40
35
30
1.5
2.0
2.5
3.0
3.5
T (20 + Log t) x 10^4

Figure 34

to both creep and overspeed. The operating temperature was estimated to be 890°F. Approximately 80% of the growth of the can is attributed to creep during operation due to the martensite tempering and carbide precipitation noted along the delta ferrite stringers.

The Alnico V magnet contained there axial cracks approximately 90° apart. Two of three ran the axial length of the magnet and the third approximately one-third the length. It is believed that the cracks resulted from excessive hoop stresses caused by rotation and relaxation of the protective rotor can.

9.3 First Stage Wheel

The operating temperature of the wheel was estimated by the previously described method to be 870 and 890.

The shroud ring experienced a growth of .004 inches which was induced by creep and confirmed by the microstructure. In addition deposits up to 3.3 mils were located on the working surfaces and less than .5 mils of erosion was experienced on the leading edges.

9.4 Second Stage Wheel

Two erosion grooves were present on the I.D. surface. Erosion depth was 0.6 mils near the edge and 2.75 mils near the vanes. Deposits were also visible on the I.D. surface which had thicknesses of .85 mils. Deposits on the vanes were evident microscopically and visually. These deposits could account for the 4-5% reduction of throat area noted in post test calibrations.

The erosion grooves and deposits are the result of mercury droplets being sprayed from the blading by centrifugal force.

9.5 Second Stage Wheel

Erosion of the blades occurred at the trailing edges on the concave surface and was estimated to be 2.6 mils in depth. Deposits up to 0.8 mils were also noted on the working surfaces but no evidence of corrosion was found. The estimated temperature of operation was 750°F.

9.6 Third Stage Nozzle

Similar erosion grooves were observed on the I.D. of the nozzle. These grooves were 2.7 mils in depth. Deposits 2.3 mils thick were also noted. Corrosion products in the vane region had the following spectrographic analysis:

Major - Co, Cu

Appreciable - Si

Minor - Cr, Fe, Ni

Trace - Mo, Mn

Total reduction of area was 4-5% as determined from post test nozzle calibration.

9.7 Alternator Bearing Sleeve

The bearing sleeve showed scoring, galling and seizure. A crack was also apparent on the journal, running in an axial direction. Microscopic examination of the journal revealed the following:

a. Localized corrosion and surface damage up to .85 mils.

b. A heat effected white layer approximately 1.6 mils thick with some radial surface cracking and an overall heat affected zone up to 7 mils deep. This damage was apparently caused by the seizure and rubbing which occurred during shut down. The white layer suggests contact temperatures in excess of 2000°F.

9.8 Pump Impeller

The pump impeller suffered cavitation erosion damage on the body and blades. The damage is similar to that experienced in previous endurance testing. Removal of material from the inlet of the blades was ragged and suggests that small particles were dislodged during operation. The maximum depth of the erosion was about 25 mils.

9.9 Electrical Connector

The electrical connector which shorted (test rig component) was sectioned and microscopically examined. The following factors were noted:

a. The solder appeared to have melted and partially run out.

b. Oxidation of the copper were strands occurred.

c. Some plating remained on the connector surface

The solder used in the joint was 95% Sn-5% Sb with a melting point of 450°F. Actual operating temperature of the connector was 350°F prior to failure. However, metallurgical inspection indicates that at temperatures above 250°F, mutual diffusion of the copper and solder results in the formation of brittle copper tin compounds. As a result the joint could fail rather easily if movement is encountered. It is difficult to pinpoint the exact cause of failure but it is believed that mechanical failure occurred which resulted in electrical arcing which subsequently caused the overheating, solder melting, shorting, and alternator rotor demagnetization. It is suggested that higher operating temperature solders (such as silver solder) be employed in the test rig components.

9.10 Summary

Based on the observations noted, the condition of the turboalternator was considered to be good after the 6 months of operation. It is well within the realm of reasonable speculation that the unit could have operated the entire one year period particularly in light of the fact that no deterioration in performance was experienced.

However, the examinations of the components as described in the previous text indicates areas of improvement and some potential problem areas. These are:

a. The pump impeller - it is suggested a harder material such as AISI T1 or M4 tool steel be incorporated plus more positive action taken to insure no air leakage into the impeller from the test rig.

b. First stage turbine wheel shroud ring - this material should be changed to afford higher creep strength properties than the PH 15-7 Mo. An alternative suggestion is AISI H 11 alloy steel.

c. The rotor alternator can - similarly this material may require higher creep strength if the rotor can not be maintained below approximately 850°F. An alternative solution would be a slightly greater thickness of the shrink can.

References

1. Evaluation of Sunflower CSU I-3A Turboalternator Components After Six Months of Operation - TRW TM-3893

10.0 CONCLUSIONS

The results of CSU I-3A turboalternator testing support the conclusions reached after previous Sunflower CSU tests. The only instance in which failure of the unit could be attributed to the unit itself was the very first test (CSU I-1-1). Certain design deficiencies were apparent in that unit and after modifications based on the results of the CSU I-1-1 test were incorporated into subsequent units, no test failure occurred as a direct result of a turboalternator malfunction. This indicates that the basic design of the Sunflower turboalternator is sound and with a few refinements the CSU can logically be expected to achieve the objective of one year's continuous operation of full power output.

Performance analysis of the units tested to date indicate that, with no manufacturing deviations, the present unit can supply the full useful power output of 3000 watts even with a 10% degradations in performance. Metallurgical inspection and analysis of CSU I-3A hardware has indicated that the unit could reasonably have been expected to operate for the full year design objective. This capability can be even more assured by incorporating certain design refinements which will be discussed in a later portion of this section.

Figure 35 is a summary of Sunflower CSU turboalternator testing to date. The total operating time for the seven test runs is 6792 hours and, as previously mentioned, only the CSU I-1 test was terminated because of turboalternator malfunction. The rest of the tests were shut down for various reasons as follows:

a) CSU I-3-1 because of corrosion products blocking the first stage nozzle.
b) CSU I-1A-1 was voluntarily shut down after completion of test objectives.

c) CSU I-1A (in PCS I-1) because of a system malfunction.

d) CSU I-3A-1 because of corrosion products from the rig partially blocking the first stage nozzle.

e) CSU I-3A-2 because of a municipal power failure.

f) CSU I-3A-3 because of a test rig electrical connector short.

The developmental history section described the design evolution from CSU I-1 to CSU I-3A and comparison of the test results for these two units shows the improvement in performance that has been achieved. There are additional refinements in the design that can be incorporated, however, to further improve performance and assure the capability of one year's operation. These are as follows:

10.1 Turbine Nozzle Throat Areas

In no case has a CSU been tested that had all three turbine nozzle areas at their design values. No correction has been made on the first or second stages but the third stage area for all units assembled after CSU I-1A has been corrected by blocking the appropriate number of passages. By resizing the blade height and by more accurately controlling the manufacture of the nozzles, design areas can be obtained. This will improve turbine efficiency and reduce erosion.

10.2 Turbine Inlet Vapor Scroll

The 360° turbine inlet used on all units except CSU I-1 caused excessive heat to be removed from the vapor stream before entering the first stage nozzles. This heat removal created or contributed

TURBOALTERNATOR TEST SUMMARY

Unit	Test No.	Test Objectives	Duration Hours	Reason for Shutdown	Corrective Action & Disposition of Unit
CSU 1-1	1	Performance Checkout	1.5	Bearing Wear and Improper Package Drainage	Unit Redesigned and Rebuilt
CSU 1-3	1	Performance Checkout and Endurance	2348	Obstruction of First Stage Nozzle with Products of Corrosion	Modification of Test Rig and Redesign of Turbine Inlet Line. Unit Rebuilt into CSU I-3A. Alternator Thrust and Journal Bearings Unchanged
CSU 1-1A	1	Performance Checkout	64	To Integrate with System	Installed in System
	2	System Performance Checkout	50	Test Stopped by System Malfunction	Cleaned and Stored
CSU 1-3A	1	Performance Testing and Endurance	769	Foreign Particles Dislodged from Test Rig Resulting in Partial First Stage Nozzle Blockage	First Stage Nozzle Cleaned and Unit Reassembled
	2	Endurance	3556	Municipal Power Failure	Restart
	3	Endurance	4.0	Test Rig Electrical Connector Short leading to Unit Overspeed and Seizure	Unit Disassembled for Inspection and Analysis

NOTES: 1. Alternator and thrust bearing set used in CSU I-3 and CSU I-3A total cumulative test time 6677 hrs. (9.2 Mo.)

2. No detectable change in CSU I-3A output power as observed in CSU 1-3 indicating nozzle obstruction problem corrected

FIGURE 35

to the following problems:

a) Metallic deposits within the turbine (during CSU I-3A-1) which plugged nozzle throats causing reduced performance.
b) Condensation in the scroll.
c) Reduced turbine inlet enthalpy causing reduced performance from an ideal energy viewpoint.
d) Reduced interstage quality which caused reduced performance (increased moisture drag) and contributed to blade erosion on the 2nd and 3rd stage wheels.

These problems can be eliminated by the return to the direct entry (ducted) turbine inlet vapor scroll. CSU I-2A incorporates this change.

10.3 Turbine Erosion (General)

Turbine erosion should be greatly reduced, if not eliminated, by the changes cited in paragraphs 10.1 and 10.2. An additional occurence of erosion, however, are grooves in the I.D. of the second and third stage nozzles. These are caused by the centrifugal action of the turbine wheels on liquid mercury trapped in the wheel races. This will be partially corrected by reduction of the moisture content of the vapor as cited in paragraph 10.2 and can be eliminated by the addition of drain grooves referenced to turbine exhaust. Drain grooves in the third stage wheel race have been incorporated in CSU I-2A.

10.4 Pump Performance

Pump performance can be improved and erosion decreased by redesigning the blade profile to a more optimum form and by increasing the number of back face vanes to reduce leakage. The impeller in CSU I-2A incorporates both these modifications. In addition, fully welded test rig plumbing at the pump inlet will reduce the possibility of air leakage into the impeller.

10.5 Screw Pump Seal

The screw pump seal between the alternator bearing and alternator rotor cavity was ineffective at normal bearing pressures and design speed. This allowed liquid Hg leakage into the alternator rotor gap, which caused drag and erosion on the rotor and may have been responsible for a small crack in the alternator stator bore seal. A further detriment of such leakage is the adverse effect on prototype condenser operation. Screw seal leakage into the alternator rotor cavity eventually is routed the turbine exhaust as a liquid and could cause a liquid-vapor weight ratio in excess of acceptable limits for the condenser.

A new screw pump seal has been designed and tested and found to perform excellently for the requirements of Sunflower. The new seal occupies about the same space as the old seal. The major design changes were placement of the threads in the seal I.D. (stationary member) instead of on the shaft and increasing of the number of thread leads. CSU I-4 and CSU I-2A incorporate this new seal.

10.6 Turbine Bearing Temperature

Temperatures in the general vicinity of the turbine bearing were excessive throughout the test. While this condition did not hurt bearing performance, it created undesirable effects such as very light corrosion of the journal sleeve and a saturated liquid vapor bearing drain condition where temperatures were sensitive to drain pressure. Heat in the bearing area comes primarily from four sources:

a) From conduction along the bell shaped housing into the bearing bushing support.
b) Through the shaft into the journal sleeve.
c) Vapor leakage into the turbine bearing drain from the labyrinth seal by-pass cavity.
d) Contact of the inboard bearing lube drain fluid with the separator or heat shield which is exposed to high temperature labyrinth seal vapor leakage on the other side.

Heat source item (c) above will be eliminated by application of a screw pump seal to form a stable liquid vapor interface and prevent entry of vapor into the bearing drain area. Heat source item (d) above will be minimized by a double walled heat shield with its inner cavity referenced to the bearing drain environment. Both these changes have been incorporated in CSU I-2A.

10.7 Alternator Cooling Jacket

The test results demonstrated a significant gradient (over 100°F) across the alternator housing from one end of the alternator stator to the other. This gradient did not appear to affect alternator performance, but was undesirable in terms of internal temperature limits. Maximum surface temperatures in the rotor-stator gap are established by the alternator design. On the other hand, minimum surface temperatures must be high enough to prevent adverse effects from Hg condensation in the gap. Therefore, an excess gradient along the length of the gap could force internal temperatures approaching both limits, demanding precise coolant temperatures. Flattening of the axial temperature gradient will provide greater latitude for control and optimization of gap temperatures with alternator coolant. The gradient is primarily forced by the heat source effect from the turbine exhaust and the heat sink effect of the alternator-thrust support housing relative to the alternator stator. The CSU I-3A coolant jacket was incapable of counteracting this heat source-sink effect due to its parallel flow path construction.

The coolant jacket of CSU I-2A has therefore been replaced with a spiral wound coil of tubing brazed to the alternator housing. Coolant will enter the coil at the turbine end of the alternator stator and leave at the opposite end, essentially in counter flow with the imposed gradient. The net effect will be to reduce the housing axial temperature gradient and therefore also the axial gradient existing along the rotor-stator gap.

As shown by the foregoing, all of the above items have been incorporated in CSU I-2A with one exception. The three turbine nozzles are undersize and this will result in a low power output. Based on the results of the CSU I-3A test series this unit should be capable of operating for a full year barring unforeseen occurrences such as power failures or rig malfunctions.

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